Marine Engineering

Vol. X. FEBRUARY, 1905. No 2.

TRIAL TRIP OF THE CHATTANOOGA.

The United States cruiser *Chattanooga* exceeded her contract speed of 16.5 knots on her official trial off Cape Ann, Mass., December 31, making an average of 16.605 knots in a continuous run of four hours in open water. The trial board were enthusiastic in their praise of the boat. The wind blew lightly from the west, and the sea was comparatively flat, the general weather conditions being nearly perfect. Her engines worked evenly and on the whole perfectly throughout. The *Chattanooga* was next put through an endurance test of 24 hours, returning to Newport, R. I., January 4, in a severe storm, which tested the capabilities of the vessel in a manner not down on the trial programme, and the vessel proved herself a staunch sea boat.

THE CRUISER CHATTANOOGA.

[Photograph copyright by E. Muller.]

The *Chattanooga* is a sheathed semi-protected cruiser of 3,200 tons displacement, 292 feet long, 44 feet wide, and drawing 15 feet 9 inches. She is one of the six ships known as the *Denver* class, the other four being the *Galveston, Tacoma, Cleveland,* and *Des Moines,* and is the last of the six to be completed. She is fitted with twin-screw, vertical triple-expansion engines designed to develop 4,700 horse power, and to give her a speed of 16 1-2 knots. Each engine has four cylinders, with diameters respectively 18, 29, 35 1-2, and 35 1-2 inches, and a common stroke of 30 inches. Steam is generated by six Babcock and Wilcox water-tube boilers with an aggregate grate area of 300 square feet and a total heating surface of 13,200 square feet, the ratio of heating surface to grate area being thus 44 to 1. The total weight of all machinery, with water in the boilers and condensers, is 405 tons. The normal coal supply is 467 tons, but bunker capacity is provided for a total of 800 tons, which, at a speed of ten knots,

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is estimated to provide a steaming radius of 6,925 nautical miles. The main battery consists of ten 5-inch rapid-firing guns; the secondary battery of eight 6-pounders, two 1-pounders, four Colt automatic guns, and one 3-inch field gun. The crew numbers 19 officers and 308 men. There is a protective deck which measures 2 inches on the thickest part of the slopes and 1 inch on the balance.

The *Chattanooga* was authorized by act of congress approved March 3, 1899, and a contract was entered into on December 14 of that year, with the Crescent Shipyard at Elizabethport, N. J., the price of hull and machinery to be $1,039,966. The keel was laid on the 29th of March, 1900, and after many delays the ship was launched on March 7, 1903, her contract date of completion having been June 14, 1902. When the shipbuilding trust, of which the Crescent Shipyard was a member, collapsed, the vessel was taken possession of by the government, and on September 15, 1903, the contract was declared forfeited. A month later the ship was removed to the government yard at New York for completion, it being estimated at the end of the last fiscal year that the hull was about 89 per cent. finished.

MARINE ENGINEERING DATA CARD. FEBRUARY, 1905.

NAME—Chattanooga. NATIONALITY—American. TYPE—Semi-protected cruiser.

LAUNCHED—1903. BUILDERS—Crescent Shipyard and United States Navy Yard, Brooklyn. OWNER—United States Navy.

Length on water line	292 ft. 0 ins.	Total heating surface	13,200 sq. ft.
Breadth, molded	44 " 0 "	Ratio	44 to 1
Draught, mean	15 " 9 "	Horsepower per square foot of grate	15.67
Displacement	3,200 tons	Square feet of heating surface per horsepower	2.81
Block coefficient	0.554	Indicated horsepower, designed	4,700
Engines: Two, four-cylinder triple expansion		Total machinery weights	495 tons
Cylinders	18, 29, (2) 35 1-2 ins.	Horsepower per ton of machinery	11.6
Stroke	30 "	Speed in knots, designed	16.5
Boilers:		Admiralty coefficient	207
Six Babcock and Wilcox water-tube.		Normal coal supply	467 tons
Total grate area	300 sq. ft.	Total bunker capacity	800 "

A Turbine Steamer.—The following lucid statement appears in a recent issue of the *Baltimore News:*

"To Editor Woman's Inquiry Column:

"What are the specifications of a turbine steamer?

"IGNORANCE."

"I suppose you mean what are the distinctive features of a turbine steamer. There is only one important difference between a turbine and a propeller. The former is a side-wheeler, the latter has the wheel in the rear of the boat."

PRACTICAL POINTS ABOUT THE SCREW PROPELLER.

BY W. F. DURAND.

PART II.

LAYING DOWN A SCREW PROPELLER ON THE DRAWING BOARD.

Having considered in the preceding chapter the various types and forms which the screw propeller may take, we come next to the problem of the delineation or laying down of the propeller on the drawing board. We assume for the first example a propeller of simple type, solid hub and blades, uniform pitch, and with blades at right angles to the axis and symmetrical in form about the blade center line. Now, it must be remembered, that in all cases of propeller delineation the following items are indispensable to the molder or pattern maker:

(1) Information serving to determine the area and contour of the blade.

(2) Information serving to determine the pitch.

(3) Information serving to determine the blade thickness and its distribution.

(4) Information serving to determine the hub.

Any information beyond that needed for these purposes is, as a rule, quite unnecessary, though occasionally the various projected views may be of value with reference to questions of clearance relative to stern post or arch of propeller well. Having in view then the above schedule of information, we may proceed as follows:

In Fig. 16 draw XY to denote the shaft center line, and OU a line at right angles with it, on which lay off a distance OC to represent the radius. Then lay off the diameter and length of the hub and put in the hub contour as shown. This is to be considered only a first approximation to the hub, especially as regards length, which may be modified later as the blade dimensions may require. We are now ready to determine No. (1), the blade area and contour.

The area numerically has been specified by the designer so that its amount is known in square feet. Suppose that we are dealing with a three-bladed propeller; then, since the blades are symmetrical about the blade center line OC, the area to be inclosed by the half blade contour $FCDE$ is known in square feet as one-sixth the total amount. We then divide this area by the distance FC or length of the blade in feet. The quotient will give the width of a rectangle of area equal to that of the half blade desired. Lay off this width, represented by CG, and draw GH. Then the rectangle $FCGH$ contains the area of one-half a blade, and a contour CDE may be sketched in by the eye in such way as to contain practically an equal area, and at the same time to have the general form of contour desired. The use of the rectangle is therefore purely as an aid in sketching in free hand the half contour in such way that the area may be closely realized and the contour determined in accordance with the taste of the designer. When satisfied on these points the curve may be smoothed out and taken as a trial contour. The accuracy of the area may now be tested, if desired, by running a planimeter about the contour and comparing with the desired amount. When thus adjusted to the satisfaction of the designer the contour is duplicated on the other side of the blade center line FC, and the entire contour is determined.

Instead of determining the contour of the blade in this manner by judgment, and with reference primarily to a rectangle as a starting point, it may be desired to develop an elliptical contour. To this end we may proceed as shown in Fig. 17. Let OC be the radius of the propeller as before. Then from A, the middle point of OC, describe the semi-circle CBO. Then lay in the hub contour as before, and divide the length of blade FC into any convenient number of parts and draw lines parallel to XY as shown. Then if distances PH_1, AB_1, LD_1, etc., are laid off as a fixed fraction of the corresponding ordinates of the semi-circle PH, AB, LD, etc., the points H_1, B_1, D_1, and others similarly determined will lie on an elliptical contour. Thus, if the fraction is .5, for example, we shall have $PH_1 = .5PH$, $AB_1 = .5AB$, $LD_1 = .5LD$,

and similarly for all other like ordinates. In this manner, after determining the fraction which is to be used in any given case, the correct elliptical half contour may be readily determined, and then doubled on the other side of the axis *FC*. It only remains to give a rule for determining approximately the fraction to be used in any given case. The exact value will depend on the

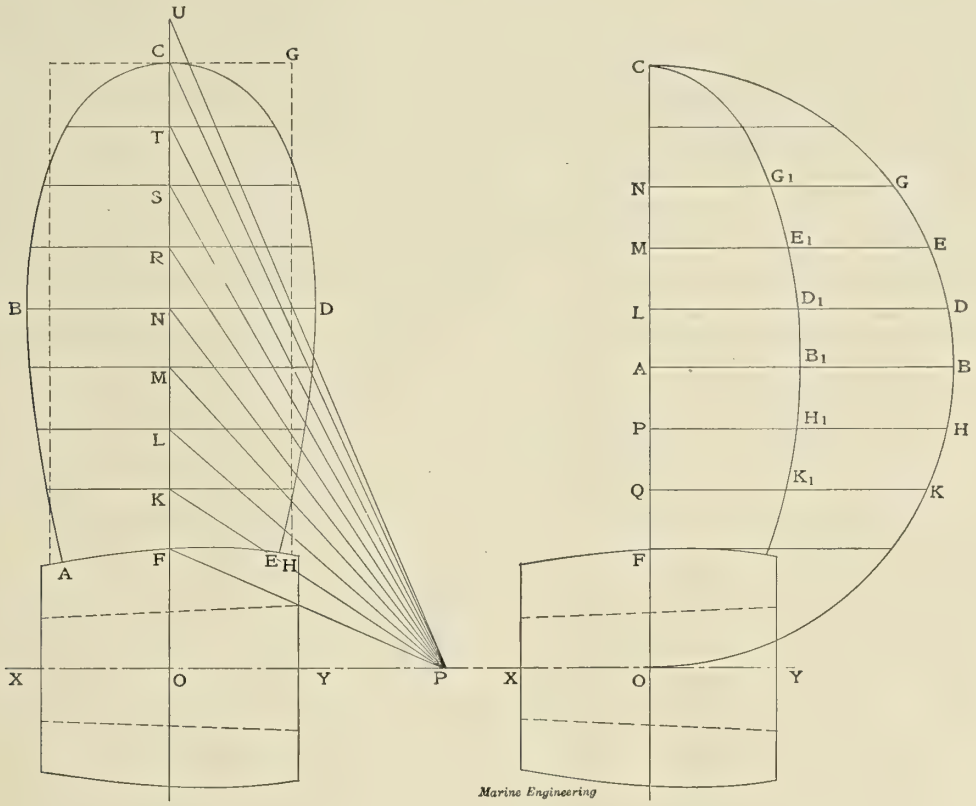


FIG. 16. FIG. 17.

diameter of the hub; but for ordinary cases where the hub is about .20 the diameter of the propeller, the following rule may be employed:

Rule: Multiply the area of *one* blade by 5.66 and divide by the square of the diameter. The quotient will be the fraction suitable for fairly large size solid hubs. For small solid hubs the correct value will be slightly less, and for large hubs with detachable blades, its value will be slightly greater.

In any case the contour thus determined may be tested by planimeter and corrected as found necessary.

It should be understood, of course, that there is no especial virtue in the ellipse as a form for a propeller blade, and there is no reason for supposing that such a contour is better than any other generally similar form. It does, however, give a well formed blade, and has often been preferred by designers as a standard form, or in some cases as an initial form which may then be modified by taste or judgment as desired.

We now turn to item No. (2), the determination of pitch. It is seen from Fig. 1* that the pitch at any given small element will depend simply on the angle *APC*, which the plane of the element makes with the plane of rotation, and for a uniform pitch propeller this angle will be the same for all points at the same distance from the axis of the propeller, but will naturally vary with the distance from the axis. This angle for any particular radius is determined, furthermore, as shown in Figs. 3* and 18 at

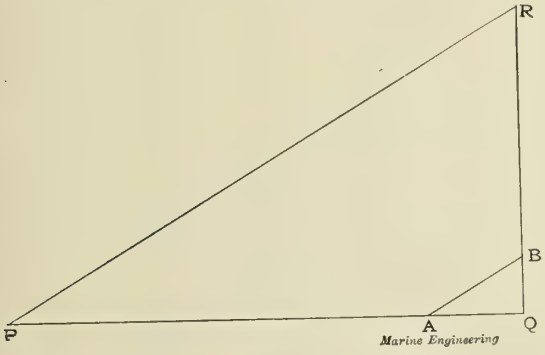


FIG. 18.

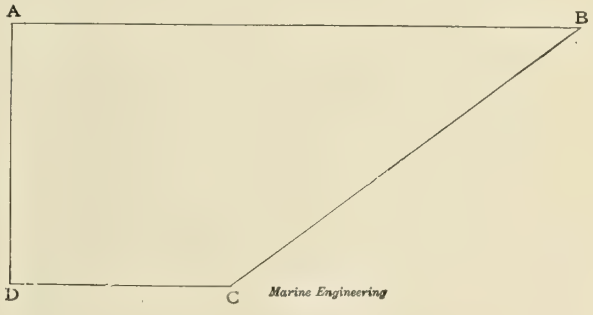


FIG. 19.

QPR, as the angle of a right triangle of which the sides about the right angle are, one *PQ* the circumference traced by the point *P* in a complete turn, and the other *QR* the pitch. If, therefore, we lay off a right angle triangle with these two sides about the right angle, then the angle opposite the pitch will be the pitch

*See MARINE ENGINEERING, page 14, January, 1905.

angle desired. Now, if the pattern is to be made in wood, it may be desired to know this pitch angle at a considerable number of distances out from the shaft center, so that the slope of the blade may be correctly determined at a corresponding series of localities between hub and tip. This might therefore be done by determining such a triangle for each distance from the shaft axis. Suppose, however, we should divide both sides, PQ and RQ, by

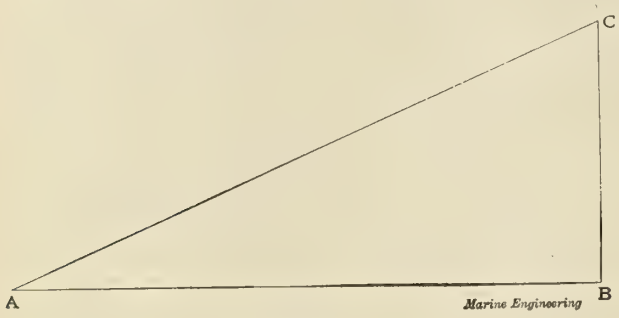


FIG. 20.

2π, or 6.2832; we should then have distances AQ and BQ reduced in this proportion, but determining the same angle $QAB = QPR$. Of these two sides it is seen that $AQ = PQ \div 2\pi$ = circumference $\div 2\pi$ = radius, while BQ = pitch $\div 2\pi$. If, therefore, we should lay off a right angle with two sides, one the radius and the other the pitch $\div$ 6.2832, the angle opposite the latter side

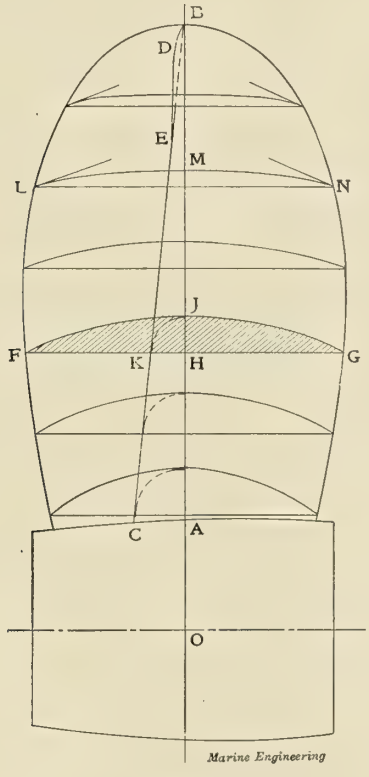


FIG. 21.

will be the pitch angle desired. This is actually carried out as follows:

In Fig. 16 lay off OP equal to pitch $\div$ 6.2832. Then from M, any point at which it is desired to determine the pitch angle, draw MP. Then the two sides of the triangle formed are MO and OP, and the angle PMO must therefore represent the pitch angle desired. If, then, a series of such lines be drawn to points F, K, L, M, etc., the lines PF, PK, PL, PM, etc., will give by their inclinations to the radius OC the values of the pitch angles at these various distances from the center line.

These angles are useful to the pattern maker for testing the slope across his pattern at any given distance from the center line. They may be employed in the following manner: Let a piece of sheet metal be cut in form as shown in Fig. 19, the angle at B being made equal to PSO, for example, in Fig. 16. Then suppose this strip of metal bent or rolled to a circular arc whose radius is OS. The strip thus bent will then form part of a cylinder of this radius, and a comparison of this with the construction of Fig. 2* will show that the line CB, Fig. 19, is similar to PR of Fig. 3, and therefore when rolled up to fit the form of the cylinder the edge CD will come exactly to the helix for this particular pitch and radius. Placed on edge, therefore, with CB downward and with DC resting on its circular arc of radius OS, the edge CB will serve as a test for the face of the pattern which, face upward, may be swung under CB and examined as to the closeness of fit. A series of such guides will thus serve to determine a series of lines across the blade, each of which will lie on the helicoidal surface which is to form the face of the blade. Such a series of lines or grooves across the pattern block being determined, the remaining material is readily brought down to a

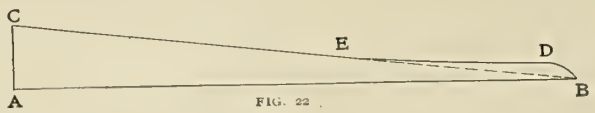


FIG. 22.

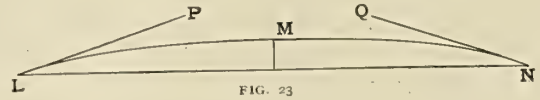


FIG. 23.

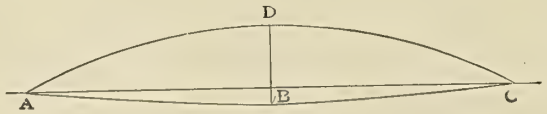


FIG. 24.

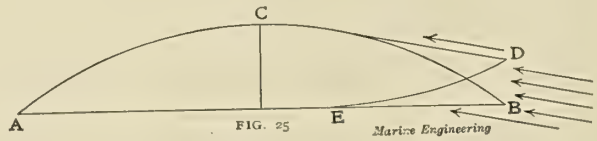


FIG. 25.

smooth and continuous surface, giving a very close approach to the true helicoidal surface desired.

If, on the other hand, the blade is to be swept up in the foundry, there will be needed but one such guide, commonly known as a *guide iron*, and cut to the angle which would be given by a line drawn to some point (U, Fig. 1), lying just clear of the tip of the blade. In this case the guide iron is cut triangular in shape, as in Fig. 20, the angle BAC representing the inclination of PU to the radius, and therefore giving the pitch angle at this point on the surface supposed to be extended just beyond the tip of the blade. If next this guide iron is rolled into the arc of a circle of radius OU and stood on edge, it will serve as a guide to a horizontal sweep working on a vertical axis, and the edge of the sweep thus guided will by its movement generate the desired helicoidal surface, and the mold for the blades may thus be formed in the foundry and without the use of a pattern in wood.

We now turn to the last important feature which is to be shown on the drawing—the thickness and its distribution.

*See Marine Engineering, page 14, January, 1905.

Rules for the determination of the thickness at the hub will be given at a later point. In Fig. 21 let *AC* be the midrib thickness at the hub, as thus determined by rule. Then draw a line *CB*. This would give a suitable midrib section from root to tip except that at the latter point it would become too thin for casting. To remedy this we modify the thickness at the tip as follows: At *B* lay off a curve *BD* with slope at *B* representing about the sharpest angle which can be readily cast with work of this character. Then run this curve into *CB*, with a reverse curve *DE*, and the section *ACEDB* may then be taken as a suitable midrib section or thickness layout from root to tip of blade, giving by its width at any given point the mid thickness for the transverse section of the blade at this location. The construction of this midrib section is shown in larger scale in Fig. 22. Then to lay out the thickness section at any location as *FG*, the thickness *HK* is taken as radius and a quadrant *KJ* is swept over to the point *J*. Then a circular arc drawn through *F*, *J* and *G* will give for this location on the blade the form of section as shown.

For the thin sections toward the tip of the blade, this operation will require some modification. Here a circular arc passed through three points such as *L*, *M* and *N* would give edges altogether too sharp at *L* and *N*. To correct this we may draw through *L* and *N* lines *LP* and *NQ*, Fig. 23, making with *LN* the minimum angle which can be readily cast, and varying from 8 or 10 degrees for bronze to 20 or 30 degrees for cast iron. A curve may then be run in as shown, tangent to *LP* and *QN* at the ends, and parallel to *LN*, passing through *M* at the center. This will then give a suitable thickness section which will not be too thin at the edges.

In some cases a part of the thickness is put on the driving face, though this is a practice which can scarcely be recommended. This layout of thickness is sometimes adopted for sections near the root of detachable blades in order to give a better arrangement of bolt holes, but it is quite sure that such distribution of thickness will entail some loss of efficiency. In such case the total thickness, *BD*, Fig. 24, is divided as shown, and placed partly on the face and partly on the back, giving to the face a slightly rounded form.

Another modification sometimes met with, and which is presumably less objectionable or even probably beneficial, consists in modifying the sections on the leading edge near the root, as shown in Fig. 25. The section *ABC* shows the form which would result with the usual mode of procedure as already described. Where the sections are thick, however, it is quite likely that the flow of the water to the propeller may be in such direction as to encounter in part the back of the blade instead of the face. This would be the case for a relative motion of water and blade in the direction indicated by the arrows. In such case the back of the blade would be subjected to a direct force developed by this motion, giving a component directed aft and opposed to the movement of the boat. To avoid this result the sections may be modified as shown by *ACDE*, bringing the developed forces all on the face side of the blade, and with components forward instead of aft. In any event this modification of the section is only significant for the thicker parts of the blade near the root, and usually need not extend beyond from one-third to one-half the blade length from the root toward the tip.

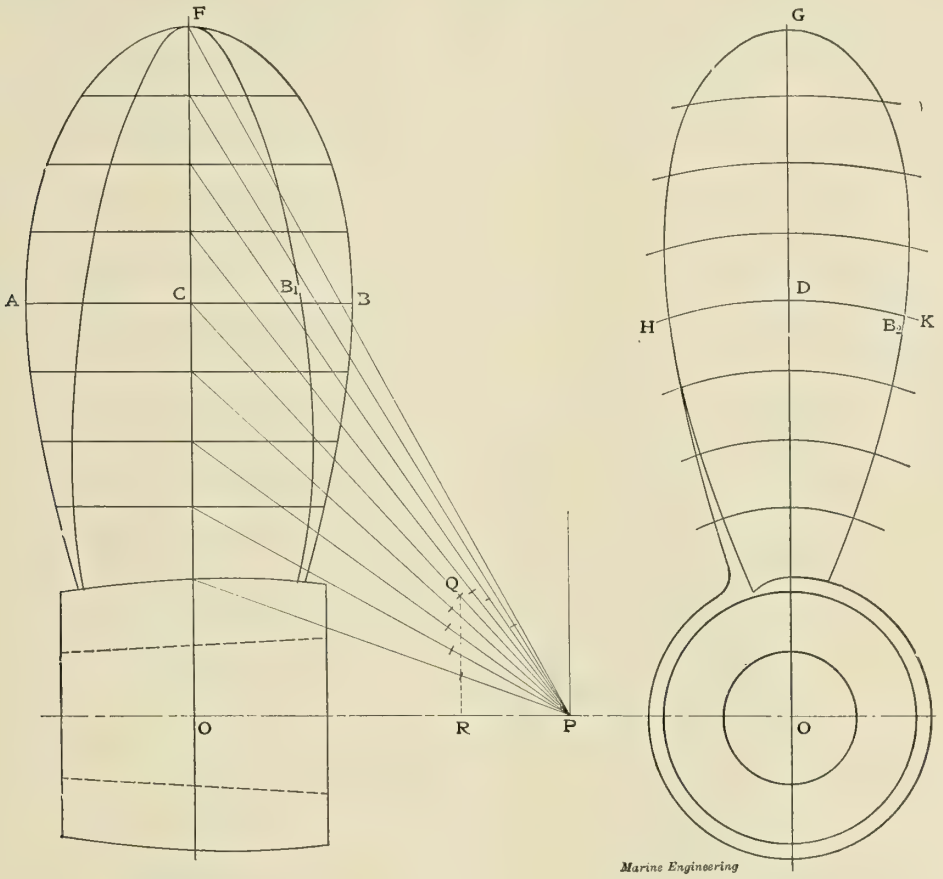


FIG. 26a. FIG. 27a.

Projected Form of Blades. We must next consider briefly the simplest method of obtaining an approximate projection of the blade contour on a plane either parallel to the axis or at right angles to it. It must be understood in advance that the methods in common use are not rigorously exact, as such methods would be very tedious in application. They are, however, quite sufficiently exact for all engineering purposes, and are employed practically rather than more complex methods.

In Fig. 26 (a) let the outer blade contour represent the actual or so-called developed blade form. Then we may conveniently split up the operation in question into a series of steps as follows:

(1) Lay down the end view of the hub as shown in (b) and draw a blade center line OG representing the line OF in (a).

(2) Lay off OP = pitch ÷ 6.2832, thus determining the point P.

(3) Let AB be any breadth across the blade. Then draw CP giving the angle PCO as the inclination of the element to the transverse plane, and CPO as the inclination to the longitudinal plane.

(4) Lay off $PQ = BC$ and draw QR as shown.

(5) Then PR is taken as the projection of CB on a longitudinal plane, and is laid off as CB_1 thus giving the point B_1 which is taken as the projection of B on a longitudinal plane.

(6) Next project C over to D and draw an indefinite arc HK.

(7) Then QR is taken as the length of a chord DB_2 thus determining a point B_2 which is taken as the projection of the point B on a transverse plane.

We have thus determined the two points, B_1 and B_2, taken as the projections of the point B of the developed contour, the former on a longitudinal and the latter on a transverse plane. In similar manner projections of other points are found, and thus the entire contour, point by point. With the developed contour symmetrical about OF as here assumed, the projected contours are likewise taken as symmetrical, one about OF and the other about OG, and the points found at B_1 and B_2 are then simply duplicated on the other side of the center line. Blades with unsymmetrical contours will be discussed at a later point.

By a further simple application of descriptive geometry we may show the end view of the blades, or of the surface which forms the driving face, but as this is rarely of practical value we shall not take the space required to describe the necessary operations.

The type of propeller thus far considered has been with uniform pitch, and blade contour symmetrical about a radial center line. We have next to note the chief varieties which may arise in the way of departures from this simple standard form.

Blades not symmetrical. In Fig. 27, let $ABCD$ represent a rectangle equal in area to the proposed blade, and determined as previously explained. Then supposing DC to be the forward edge, the designer may sketch in a contour $GHFKL$, approximately equal to it in area, but carried aft in such manner as shown in the figure. Such a contour may present somewhat the appearance of

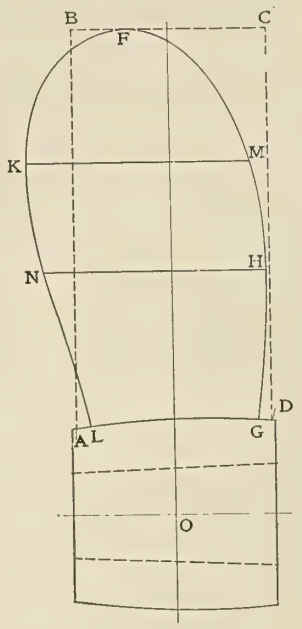


FIG. 27.

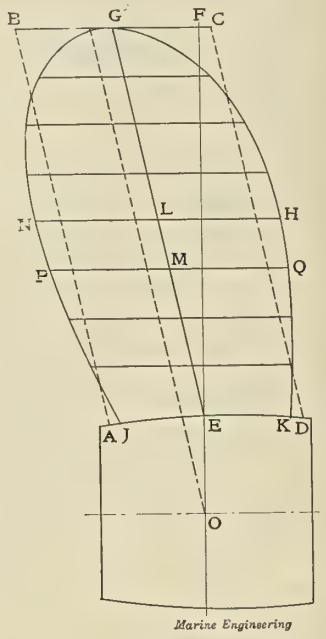


FIG. 28.

a blade bent back from the plane of rotation, as described in the preceding chapter, but quite different in the actual manner of its formation. This modification in blade contour is quite frequently met with, giving as it does, something approaching a

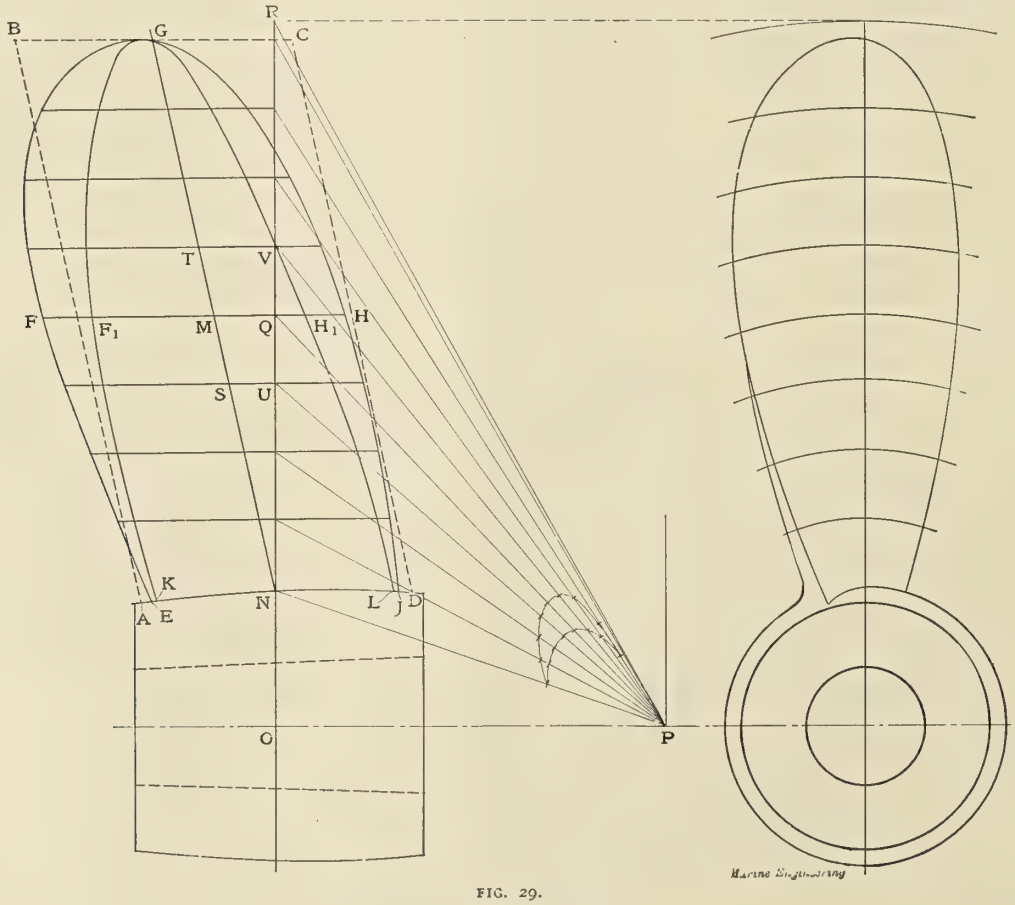


FIG. 29.

scimeter form for the leading edge of the blade as it cuts forward into the water. After thus sketching in the approximate form of blade it may be tested for area by planimeter, and corrected as may be needed to give the area specified. Whatever may be the form of contour ultimately accepted by the designer, its relation to a radial element, *OR*, will make it a simple matter to lay out such a contour on the pattern or swept mold, according to the method employed for making the propeller itself.

In order to obtain the projections of such a contour it is only necessary to treat each side of the center line separately, and in the same manner as previously described for symmetrical blades.

Blades bent aft. Let *EG*, Fig. 28, denote the line of the elements of the blade surface for this type of blade, as described in the preceding chapter, *FEG* denoting the angle by which the blade is tipped back from *EF*, the line at right angles to the shaft axis. To lay out the contour of the blade in such case we may proceed as follows: Let *GC* be determined as the half side of a rectangle of area equal to that of the proposed blade, and lay off *GB* and *GC* as shown, completing the parallelogram *ABCD*. Then the area of this parallelogram will equal that of the blade desired. The designer may next sketch in by eye the half contour *GHK* as shown, making it as nearly as possible of equal area with the half parallelogram *EGCD*. This may then be tested by planimeter, and corrected as needed, in order to give the area specified. In all such cases where the use of a planimeter is suggested for the correction of the area of a contour determined as described, by reference to a rectangle or parallelogram, it may be remarked that the performance of a propeller varies only slightly for such minor changes in area, and no serious departure from the conditions of the design is likely to result if the first approximation as determined by eye is accepted without further correction.

When satisfied with the half contour *GHK*, the half breadths, *LH*, *MQ*, etc., may be duplicated on the other side of *EG*, giving the points *N*, *P*, etc., and thus determining the remaining half contour *GNPJ*.

A blade thus determined, as shown in Fig. 28, will have equal area on either side of the line *EF*, though it will not be symmetrical about this line in the usual sense of the term.

In case it is desired to lay down a contour with unequal area on either side of the line of reference *EG*, it is simply sketched in by reference to the parallelogram in the same manner as described for the similar case with blades at right angles to the shaft axis, and as illustrated in Fig. 29.

In this general manner any form of blade may be readily specified and as readily laid down in the pattern shop or foundry.

In all such cases of blade form the main requirement is to determine the contour with reference to some line on the drawing, which line may be readily laid down on the pattern or mold. Then relative to this line the actual form is readily determined by transfer from the drawing.

The pitch angles for blade templates, guide irons, etc., for a blade tipped back as in Figs. 28 and 29, are not affected by the inclination of the blade in this manner, and are therefore the same for equal radial distances and the same pitch as for a blade at right angles with the shaft. They are therefore laid out as follows: In Fig. 29 let *M* be any point on the middle line *GN*, or *FH* any width across the blade for which the pitch angle is desired. Then we have simply to lay off $OP = p \div 6.2832$ as before, and draw *PQ*, making the right angle triangle *POQ*. The pitch angle for the element in question will then be *PQO*, the same as for a blade at right angles with the shaft and with the same pitch. Similarly for a guide iron at a radial distance *OR*, the angle will be *PRO*. In particular it should be noted, in the case of such blades, that the pitch angle lines are not to be drawn to points *S*, *M*, *T*, etc., on the blade center line, but rather to *U*, *Q*, *V*, etc., on the radial line at right angles through *O*.

The transverse and longitudinal projections of a blade thus bent aft are then found by the use of these pitch angles in the same manner as with blades at right angles with the shaft, and as illustrated in Fig. 29, similar to Fig. 26, for blades at right angles with the shaft.

Variable Pitch. We assume that the pitch is to be constant along any radial element of the blade, but varying from the leading to the following elements, with a value p_1 for the former and p_2 for the latter. Then, in Fig. 30, a lay-off, *OP*, equal to $p_1 \div 6.2832$ and *OQ* equal to $p_2 \div 6.2832$, thus giving two points,

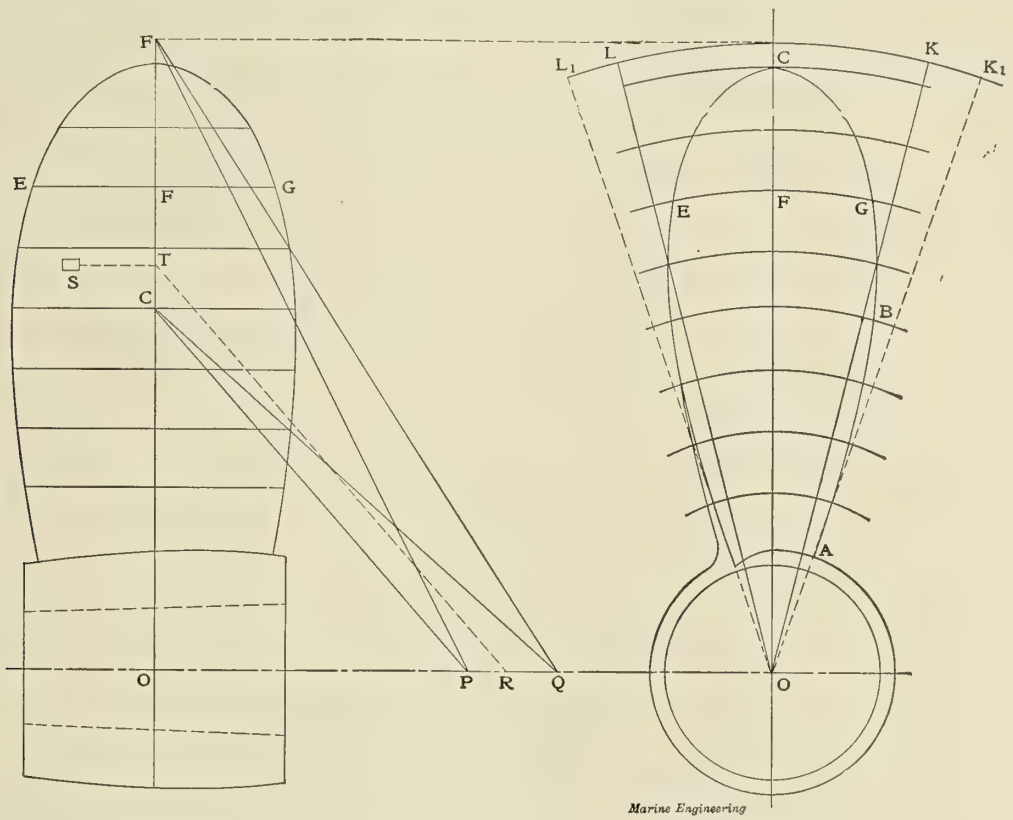


FIG. 30*a*. FIG. 30*b*.

P and *Q*, for the two values of the pitch, instead of one, as in Figs. 26 and 29. Then for any radial distance *OC* the two angles *PCO* and *QCO* give the two pitch angles or angles of inclination for a small element located in the former case on the leading edge and in the latter on the following edge. Similarly for any other radial distance on the blade, while for a guide iron beyond the tips of the blade as at *F* the angles at the two extremes of the iron are given by *PFO* and *QFO*. In like manner the angles for any intermediate blade element with its intermediate value of the pitch may be found by laying off the corresponding intermediate point between *P* and *Q*, and drawing the inclined line from any point on *OF* at which the inclination of the element is desired.

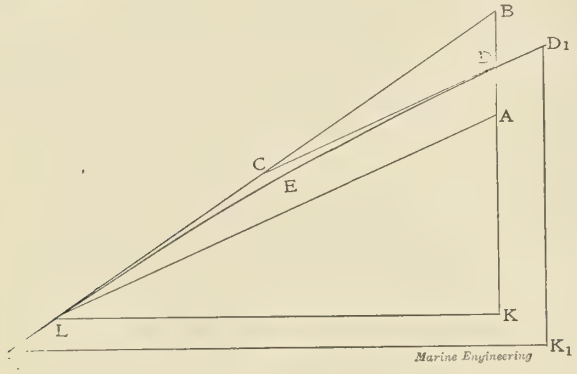


FIG. 31.

We must now consider the method of determining the projections of the developed surface in the case with variable pitch. As in other like problems dealing with the screw propeller, an approximate method will be given sufficiently accurate for practical purposes.

We have seen that the projection of a blade width depends on the pitch angle, while the latter varies from point to point across the blade with variable pitch. In order to simplify the problem, however, we may use an average value of the inclination, determined by drawing lines from a point *R* midway between *P* and *Q*, and proceed with the determination of the projected contour in the same manner as previously explained for uniform pitch. This gives, of course, only an approximation to the true form, but is usually sufficient for practical purposes, and if desired may be corrected by a method to be explained at a later point.

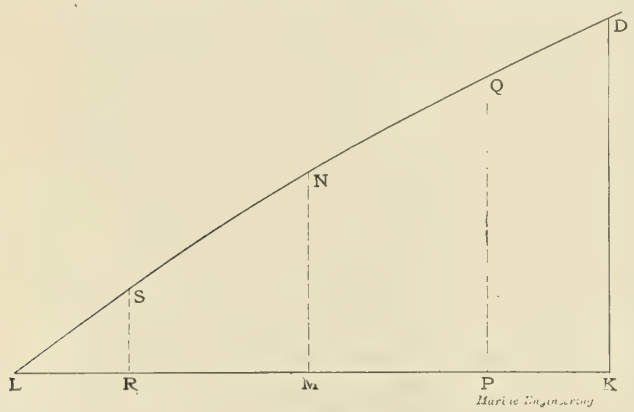


FIG. 32.

Let the projection on a transverse plane determined in this manner be represented in Fig. 30b with *ABC* as the leading edge. Then while it would be perfectly possible to give to the pitch along the curved edge *ABC* a constant value p_1, it is more simple in practice, as stated above, to select some radial line or element *OK* along which the pitch shall be p_1 and a similar line or element *OL* along which it shall be p_2. We may then proceed to develop a guide iron *LK* for such a blade in the following manner:

(1) Find the developed length of *LK*, *Fig.* 30*b*, and lay it off as shown in Fig. 31.

(2) Lay off two angles, *KLA* and *KLB*, equal respectively to *PFO* and *QFO*, Fig. 30*a*.

(3) Select by judgment a point *C*, and draw *CD* parallel to *LA*. The location of *C* should be such as to make *LC* = *CD*. It is readily seen that to this end the point *C* should lie slightly nearer *L* than *B*, and a point slightly on one side of the center of *LB* toward *L* will fulfill all requirements. A slight error in the location of this point will make no material difference with the form of the resulting guide iron.

(4) Draw a circular arc tangent to *LC* at *L*, and to *CD* at *D*, and extend beyond at each end somewhat, as shown at L_1 and D_1.

With this construction a guide iron, *LEDK*, cut out of plate and bent up as previously described, will serve to give a blade of regularly increasing pitch between the two radial elements *OK* and *OL*, Fig. 30*b*. If these elements do not contain between them the entire contour, as in the figure, then the guide iron may be extended as shown by L_1, *LE*, DD_1K_1. This will give the same law of pitch variation extended on either side of the elements *OK* and *OL*, and in this manner the surface of the entire blade may be swept up with the same general law of pitch variation. The radial lines OL_1 OK_1, Fig. 30*b*, give the extreme limits for the blade surface in this projection, and the arc L_1K_1 gives the total extent of the guide iron for the entire blade. In Fig. 31 L_1K_1 must of course be made equal to this total length of arc.

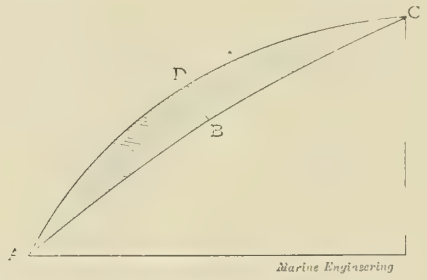


FIG. 33.

Precisely the same general method of construction may be followed for the determination of a series of templates for testing the pattern across the blade face, in case it is desired to make the pattern in this manner, and as referred to at an earlier point in connection with blades of uniform pitch.

Referring now to Fig. 32, representing such a template, we may readily indicate the method of correcting the projections previously made, should such correction be desired. Assuming that *OC*, Fig. 30*b*, lies midway between *OL* and *OK*, find *M* the middle point of *LK*, Fig. 32, and erect *MN*, thus locating *N*. Now, suppose that *LKD* in this figure is the form of the guide or template for *EG*, Fig. 30*b*. Then find the actual or developed length of *FG* and lay it off at *MP*, Fig. 32, and erect *PQ*, thus fixing the point *Q*. Similarly determine *EF*, lay it off at *MR* and erect *RS*, thus fixing the point *S*. Then *NQ* and *NS* will be the actual or developed lengths of *FG* and *FE*, Fig. 30*b*, and these developed lengths may then be compared with the original lengths in Fig. 30*a*, from which *FG* and *FE*, Fig. 30*b*, were obtained by the approximate method first used. This comparison will readily show whether any correction is needed, and if so its character and general amount.

Template forms, as shown in Fig. 32, may also be of use in laying out the transverse thickness sections.

Thus in Fig. 33, let *ABC* be the template form for any given transverse section of the blade. Then in the same manner as illustrated in Fig. 21 the maximum thickness *BD* may be determined and laid off from *B*, thus giving the point *D*. A circular arc *ADC* may then be used for the back surface of the

blade, or the general method of Fig. 23 may be used in case the circular arc would give at *A* and *C* edges too sharp for good casting. A series of such transverse blade sections may thus be laid down for the use of either the pattern maker or molder in determining the lay out of thickness over the entire blade surface.

There remains one type of blade to which brief reference may be made—that with pitch varying radially, or between root and tip of blade. The treatment of this case involves no new principles whatever, and may be readily worked out by the application of methods and principles already explained. Referring to Fig. 16 it will be remembered that the location of the point *P* will vary with different values of the pitch. If, then, this point is located for the various positions *K, L, M,* etc., along *FC,* as determined by the value of the pitch assigned to each position, and lines drawn from these points *K, L, M,* etc., each to its own location of *P,* the pitch angles will be thus determined, and the various items of the propeller may be laid out by methods of procedure entirely similar to those previously described.

These various constructions for pitch angle may be generalized as follows for any blade element with any specified pitch: In Fig. 30a let *S* denote any element whatever of the blade area, and of which the pitch is to have a specified value *p*. Then to find the pitch angle for this element lay off the distance $p \div 6.2832$, represented in the figure by *OR,* draw a line from *S* over to *OC,* the line at right angles to *OR,* and thus determine *T*. Then the line *TP* is the hypothenuse of the right triangle *TOR,* of which *TO* is the radius of the element in question and *OR* is pitch $\div$ 6.2832. The angle *RTO* is therefore the pitch angle desired. According as the pitch for the blade as a whole is uniform or variable, and if variable according to the law of variation, the application of this general method will result in the various special methods previously given.

In case the proposed propeller involves a combination of items representing departures from the simple standard type of Figs. 16 and 26, they may all be included in the treatment on the drawing board by an appropriate combination of the principles and methods explained with reference to each individual case. Thus while rarely met with we may have a propeller with the following characteristics:

(1) Blades bent back as in Fig. 28.

(2) Blades not symmetrical about any reference line, as in Figs. 27 and 29.

(3) Pitch variable axially as in Fig. 30.

(4) Pitch variable radially.

Such a case presents no difficulty whatever aside from the detailed work of carrying out the special construction which these various characteristics may require. If the fundamental methods and principles developed in connection with these various points individually are kept clearly in mind, there should be no difficulty in combining them in such manner as to care for a combination of such items in a single propeller.

In closing this chapter it should be remarked that there are some minor points of detail in connection with a propeller drawing for the use of either the pattern maker or molder which have been passed over, but which should not give trouble to any one who has clearly in hand the methods and principles discussed with reference to the main items of the work.

It should also be noted that for simplicity in the diagrams all special details of propellers with detachable blades have been omitted, in order to fix attention on the elements essential to the delineation of the propeller itself, and independent of such details as hub construction and fitting. It will be found that such omitted details relate solely to structural features, and that all problems relating to the delineation of a propeller as a combination of helicoidal blades joined to a specified form of hub may be solved by the application of the principles and methods given.

It should likewise be remembered that in all operative problems of this character, there are many ways in detail of accomplishing the same ends, especially when approximate methods are employed. Many variations in the above methods may therefore naturally suggest themselves to the thoughtful student, and many different methods for accomplishing the same result. The methods herein given have been tested in practice, are simple in use, sufficiently accurate in result for all practical engineering purposes, and involve only those items which are of actual significance for the pattern maker or molder. With such methods well in hand the draftsman should have no difficulty in giving to the pattern maker or molder the information most needed in order to produce the propeller itself, and in the form which will be most directly applicable to the special demands of the problem in hand.

FRENCH ARMORED CRUISER DUPLEIX.

French Armored Cruiser Dupleix.

Last summer there appeared in several North Atlantic ports a most striking warship flying the French flag. It was the new armored cruiser *Dupleix,* the flagship of Rear-Admiral Rivet, commander of the French Atlantic squadron. The peculiar appearance of this vessel to the eyes of one accustomed to American

and British designs is shown in the engraving herewith presented. The *Dupleix*, which was launched at Rochefort in 1900, is one of a class of three ships, and may be considered a modern representative of a type of warships which the French designers have always regarded with favor.

By the courtesy of the *Navy League Journal* the following particulars of this ship are presented: Displacement, 7,700 tons; speed, 21 knots; normal coal supply, 800 tons; maximum bunker capacity, 1,200 tons, plus liquid fuel; armament, eight 6.4-inch and four 3.9-inch r. f. guns, ten 3-pounders, and six 1-pounders. The 6.4-inch guns are mounted in pairs in four turrets, two forward and aft and one on either beam, while the 3.9-inch weapons are mounted in broadside on the gun-deck. What seems to be a tactical disadvantage, and something quite at variance with former French practice, is the fact that the ship's sides have no tumble-home, and the broadside turrets are so placed that they cannot be trained parallel with the ship's axis, thus limiting the maximum concentration of fire dead ahead or dead astern to two 6.4-inch and two 3.9-inch guns. The 6.4-inch turrets are protected by 4 inches of armor, and are furnished with armored ammunition hoists extending down to the protective deck, which is of a uniform thickness of 2 3-4 inches, and is reinforced by a continuous water-line belt, which is approximately 5 inches thick amidships and 2 inches at the extremities. This is the limit of the *Dupleix's* armor protection, except for 6 inches of steel on the conning tower; her offensive powers are supplemented by two above-water torpedo tubes located one on either broadside forward of the beam.

A glance at the following table will show that she is far from being a match for the Italian-built *Kasuga*, of almost identical displacement, which was added to the Japanese navy just previous to the outbreak of hostilities in the far east.

Name of Ship.	Displacement, Tons.	Speed, Knots.	Bunker Capacity, Tons.	Armor.			Armament.	Total Weight. One Round from all Guns.	Torpedo Tubes, above Water.
				Belt.	Sides.	Turrets.			
Kasuga	7,400	20.2	1,100	6 in.	6 in.	5½ in.	Four 8-in.; fourteen 6-in.; ten 3-in.; four small guns;	2,520 lbs.	4
Dupleix	7,700	21	1,200 + liquid fuel.	5 in.	none.	4½ in.	Eight 6.4-in.; four 3.9-in.; ten 3-prs.; six 1-prs.	953 "	2

Obviously, for obtaining the maximum of efficiency from a given displacement, the palm lies with the Italians. In fact, the *Dupleix's* design is not likely to be repeated, even by the French, for it prohibits her coping with ships of her own class similar to those which took part in the engagement of the Korea straits, and limits her province in time of war to the running down and destruction of unarmored ships which lack the battery power to keep her at a distance. In time of peace, however, her generous dimensions—426 1-2 feet length by 58 1-2 feet beam, by 24 1-4 feet mean draft—coupled with the absence of a numerous battery of heavy broadside guns, make her a roomy, comfortable vessel, and one well adapted for use as a flagship.

THE STEAM TRAWLER SWAN.

THE NORTH SEA STEAM TRAWLER.

The steam trawlers in the North sea make up as sturdy a fleet of little vessels as ever encountered the severe gales of the great oceans. The condition of the North sea is enough to discourage the heart of the bravest sailor, as rough seas are the rule and not the exception.

The steam trawler is a vessel built specially for this service, and maintains her post on the fishing grounds for weeks at a time. The boats vary much in speed and size, but the average vessel used in the North Sea service is of about 100 net tons and 270 registered tons. The length of this boat between perpendiculars is 130 feet; beam, 24 feet; and the depth 12 feet 4 inches. Three hatches are fitted forward, 3 feet 6 inches square. The bulk of the trawlers are built near Hull, England, one firm turning out about 35 of these vessels a year, costing from $25,000 to $35,000. A few are built at Selby, a nearby town. The boats are of the raised-forecastle type with machinery located aft. The bows stand high out of water with considerable sheer, bringing the deck line amidships 3 or 4 feet above the water. A poop deck is generally fitted to protect the after gang ways and the engine hatches. On the low main-deck forward all the work of the fishermen is done; namely, emptying the contents of the trawl and sorting and cleaning the fish.

The English trawl differs from the American trawls in that it is a cone-shaped net dragged over the bottom of the sea; whereas the American trawl is simply a long line with hooks attached a little distance apart and used on sailing ships.

STEAM TRAWLER LEAVING HULL.

Most of the recently built trawlers are fitted with the otter trawl, which is a decided improvement on the old beam trawl. It is a large net, shaped like a cone, with a large board fastened to each side of the oval mouth. These boards are about 6 feet high and 10 feet long, and are termed the "otter boards." On the net side of the board is bent the warp, and when the trawl is "shot" the foremost board is launched overboard first. So, when the vessel is steaming ahead, the otter board pays off, and, of course, the net follows last. When the net is out the after otter board is let overboard and the warp of this paid out; then the whole affair sinks to the bottom and drags along after the ship. The otter boards tend to spread the mouth of the net open. The size of the net is about 115 feet across the mouth and about 140 feet long. The trawl is allowed to remain down about four hours, the vessel meanwhile steaming along at full power. This renders the navigation of the vessel very difficult, as it is nearly anchored to the bottom with the weight and drag of the trawl. For this cause the trawlers have special lights so that other vessels can steer clear.

TRAWLERS UNDER REPAIR.

The several fleets of trawlers are under the direction of

admirals, who order the vessels to take a certain course to avoid fouling any of the trawls. These directions are signalled by means of flares and colored rockets. Each vessel in a fleet has to carry the special lights ordered by the North Sea Convention and the Board of Trade. They are also bound to have their letter and number on the bows, quarters, and funnels of their ships, and are forbidden to conceal them. They all shoot trawls together at a given signal from the mark boat, the signal usually being a green rocket. The trawling is done in a depth of about 23 fathoms.

A fleet consists of 40 or 50 vessels, of which there are two classes, "single boaters" and "box trawlers." These differ only in occupation, the former fish more independently while the box trawlers fish in fleets.

The single boaters take in provisions, coal, and ice for about a week; they then proceed to the fishing ground and stay there until a sufficient quantity of fish has been caught, when they return home, unload, and start off again.

The box trawler fleet consists of 60 to 70 boats, of which 40 or more are on the fishing ground at one time. The individual boat leaves home coaled and provisioned for three or four weeks, and fishes with the fleet until it is necessary again to return home for food and fuel. No ice is carried by these vessels, but they deliver their fish cleaned and packed in boxes every twenty-four hours to vessels called the "carriers." The carriers pack the fish in ice and return to port to deliver the boxes on the landing stage at the fish market. As many as 1,500 boxes of fish are taken on board of these carriers every morning. The carriers also act as mail boats, and are seldom away from port over three days. When the fishing boat returns home she usually brings in a load of fish, but otherwise all the fish are taken by the carriers.

In conclusion it should be remarked that the crews of these boats are a set of very brave and hardy men, though a large percent of them are young. They are continually saving crews of shipwrecked vessels.

STEAMER KASHING.

The Victim of a Floating Mine at Wei-Hai-Wei.

The photograph shows the damage done to the steamer *Kashing* by striking a floating mine. The vessel, which was bound from Chifu to Shanghai, struck the mine at midnight on October 25, off the Shantung promontory, and had to put back to Wei-Hai-Wei. A large rent, measuring 10 feet by 12 feet, was torn in her port bow, and the deck above was blown up. One Chinaman of the crew was killed outright, another fell through the hole in the bow and was drowned, and three were injured.—*London Graphic.*

UNIFORM SPECIFICATIONS.*

BY W. D. FORBES.

Some time since a paper was read before this association on "The Interchangeability of Units for Sea Use." The idea was favorably received. The advantage of interchangeability of parts as well as units is certainly recognized as a furtherance of engineering work.

With these facts in mind, it is now proposed to carry the idea further and discuss the advantages of interchangeability of specifications.

It must be admitted that there is no sound reason why a steam engine, for instance, should not function equally well if ordered by one person or by another, provided running conditions are the same. It would seem that if a corporation could accept an engine complying merely with certain restrictions of weight, space and revolutions, without any regard to the material used, except that it be "first class," another could accept it without going into the detail of babbitt metal, steel, or composition entering into its make-up. It is self-evident that a firm constantly bending its attention to the manufacture of a high-grade machine will understand more thoroughly what is needed in its construction than an outsider.

It will, of course, be admitted that there are certain demands which must be met in special places, as, for instance, naval or merchant marine.

Some are liberal in permitting manufacturers to use well tried material not exactly meeting specifications, and some will pass engines which do not fill the requirements as laid down in specifications, but it is held that such procedure is manifestly wrong. The function of a specification is to specify; it should not demand information which is a trade secret or which cannot be imparted without practical demonstrations and experience, or something which prohibits obtaining the desired results, as for instance that "a hard sound casting for cylinder be furnished," and exacting that no scrap metal be used, and that the mixture of iron be clearly given. No foundryman would be willing to undertake to make a close, hard, iron cylinder without scrap, and few are willing to instruct others how to make castings, and very few, to tell the truth, are able to impart the knowledge.

Some buyers are very exacting about the babbitt metal, allowing only 2 percent of lead, while others make no demand as to this metal. A babbitt with but 2 percent of lead is good for certain positions, but is not good for high-speed work, and if pounding in is required, at least 20 percent of lead should be allowed. Pounding babbitt should be resorted to only to get a solid box; making hard spots in a babbitt bearing by pounding is self-evidently a bad plan.

A pig of babbitt is supplied, and a chemical analysis shows it to meet the specifications, bearings are poured of it, and it is supposed that the desired mixture is used, but this is not so, as a chemical analysis of the babbitt from the bearing will show that the original mixture does not exist, as much of the tin has gone up the chimney in melting. Here the buyer expresses a desire for a certain mixture, and then knowingly accepts something else.

Nickel steel is an article often demanded in engine specifications; its value cannot be questioned in many cases on account of its strength, but on small forgings there seems to be endless strains set up which no amount of annealing removes. Some articles seem impossible to make of this material when of small area, and retain their shape; whether ground or turned the springing seems to continue even when no forging is done. In nickel crank shafts of about 3 inches in diameter, great difficulty is encountered by the continual spring of the forging as the several cuts are taken, and very true cranks are hardly obtainable. That crank pins must line and be true cylinders, is a perquisite of an enduring engine, and as in small engines it is quite easy to

*Paper read at the twelfth annual meeting of the Society of Naval Architects and Marine Engineers, New York, November, 1904.

design a shaft which if made of but ordinary machinery steel will stand all possible strains and service, a party who does not demand nickel steel is getting by far the more reliable articles.

Many compositions demanded by the buyers are too hard, the formulæ 88-10-2 mixture is reported to be popular mainly because it is never furnished. It is a mixture which is very difficult to machine, and for high speed bearings is never employed in commercial work. In the bearings of a wood-working machine, where lasting qualities are of great moment, nothing but babbitt will be found with never less than 20 percent of lead in it. In fact it is held that the size of bearing has far more to do with its lasting qualities than the material of which it is made.

Probably the question of what is a right and what is a left-hand engine will be a continual source of argument with many, unless the turbine comes to the rescue; but one would suppose that this could be clearly settled by each engine builder. In one specification this is to be found: "When a person is standing at the commutator end and the direction of rotation is against the clock, the plant is right-handed." While this is incorrect, it is a clear description of what the parties choose to call a right-hand engine, and one which could be furnished without further question.

The wording in some specifications is hard to understand, as for instance the following: "The lower side of the combination bed plate to be planed perpendicularly to the line of the stroke of the engine." This might mean that the bed plate is to be set on edge in the planer and its bottom planed with a down feed—all it really means is that the bed plate is to be finished on its bottom.

As to what may be properly termed "first class" in material is open to debate, but certainly anything may be properly called so which has proved satisfactory in the past, and met the requirements of strength and endurance for a long period.

That weight, space, and lasting qualities are matters of great moment, especially aboard ship, is admitted, and it is, of course, necessary to have a clear statement of what is allowable, and to know the exact conditions under which a machine is to operate either afloat or ashore, before a design can be made or a bid given. It is sometimes claimed that a bond guarantee should be furnished by a manufacturer and he be left free to select all materials and designs, and some very successful buyers do thus order and accept articles, and it is hard to see why all should not adopt the same system; but failing this, why not adopt the other system of nominating everything and having some uniform chemical or physical tests which would pass all material, no matter where it goes or for whom it is ordered. Either condition would result in work being gotten out with less loss of time and at less expense, but one or the other system should be general.

If the latter system is decided upon, it could only be with the one desire that the very best is to be obtained; no other possible motive could be ascribed to its advocates; but it places the buyer in the position of nominating what should be used and then holding those who furnish it responsible, not a tenable position; while if the former system is adopted, the desire that the best be given is far more likely to result in success, as there is then no possible division of responsibility, and the experience of those most interested would be made available.

It would certainly seem that the bonded guarantee system would present to the navy allurements. The endless detail now imposed on officers as to material would be done away with. The experience of the navy is as free as air to us and all nations, and this, coupled to commercial experience and engineering ability, must result in manufacturers turning out mechanical contrivances which will fill all requirements, as their very existence would depend on so doing.

The necessary time now required for inspection in detail would be saved, and the time question will shortly become of far greater moment than it is now, in naval construction.

A writer in the *Army and Navy Gazette* attributes the satisfactory and remarkable freedom from breakdowns in the Japanese navy in the present war, to the "liberality" shown builders in designs and selections of material; but a large factor may be the absence of the enterprising newspaper reporter.

To conclude, it is held, first, that if there is a specification for an article it should be lived up to or the article be entirely rejected, and no responsibility should rest on the maker of an article except for good work, if accepted; second, if there is no specification the maker should be held responsible *by bonds* for the satisfactory working and lasting qualities of the articles; third, that one or the other method should be adopted.

ELECTRIC AND PNEUMATIC PORTABLE TOOLS IN SHIPBUILDING.

BY FRANK C. PERKINS.

Electric and pneumatic portable tools are largely used in modern shipyards, both in this country and in Europe, each type having its own field of work to which it is best adapted. The accompanying illustration, Fig. 1, shows the operation of German electric hand drills at work on the plates of a steamship; while Fig. 2 shows an English electrically-operated portable drill, as constructed by Campbell & Isherwood, of Bootle, Liverpool, at work in a similar manner. While the pneumatic drill is largely employed, as shown in Fig. 3, the electric drill is fast supplanting it. Pneumatic tools which occupy a field exclusively their own, and with which electric tools can hardly be said to compete, are the pneumatic chipping tools and the riveter, both of which are in common use and well known. The method of operation and the auxiliary apparatus required in plate and girder structural work are well illustrated in Fig. 4, showing pneumatic riveters of the

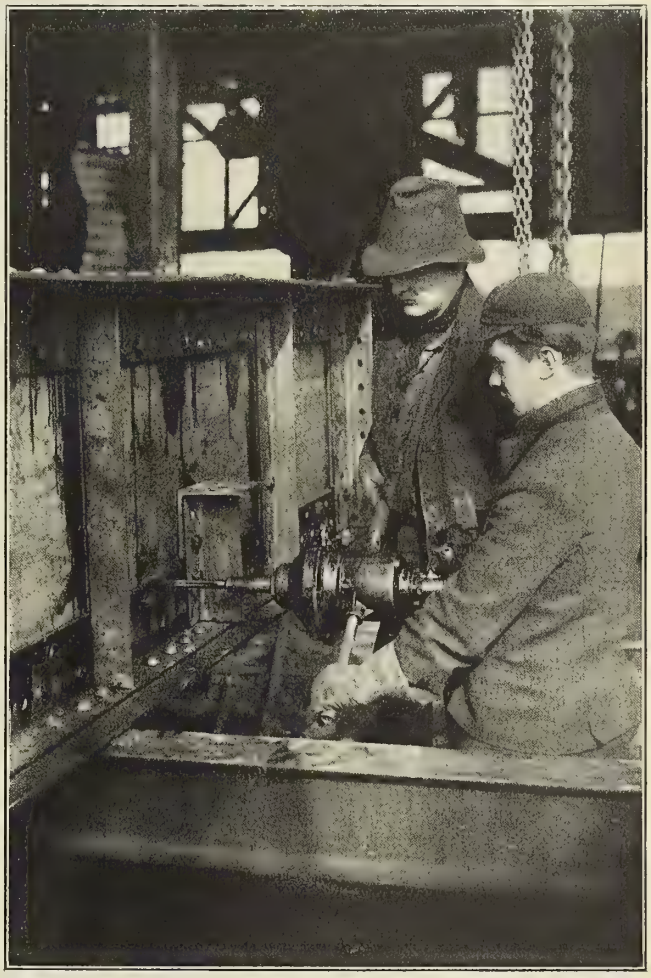


FIG. 3.—KELLER PNEUMATIC ROTARY DRILL AT WORK ON GIRDER.

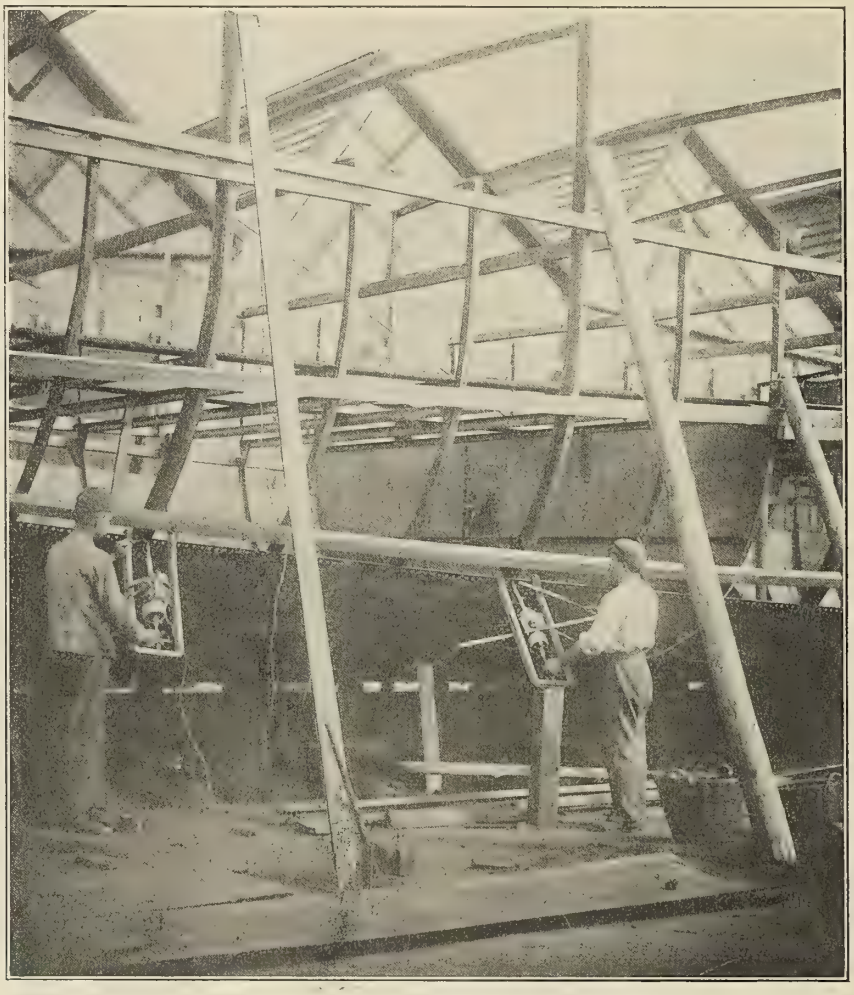


FIG. 1.—GERMAN PORTABLE ELECTRIC HAND DRILLS.

Cleveland Pneumatic Tool Company, in operation in Glasgow, Scotland.

The planing of the wooden decks of ships is one of the most laborious and unpleasant operations that fall to the lot of a ship's carpenter, and even the time-and-a-half rate allowed for this duty does not overcome the dislike for it. A large number of electric deck planers are now in operation in German and American shipyards, as well as in a number of the English shipyards on the Clyde and Tyne, which is an evidence of the alertness of the modern ship-building engineer to avail himself of the latest labor-

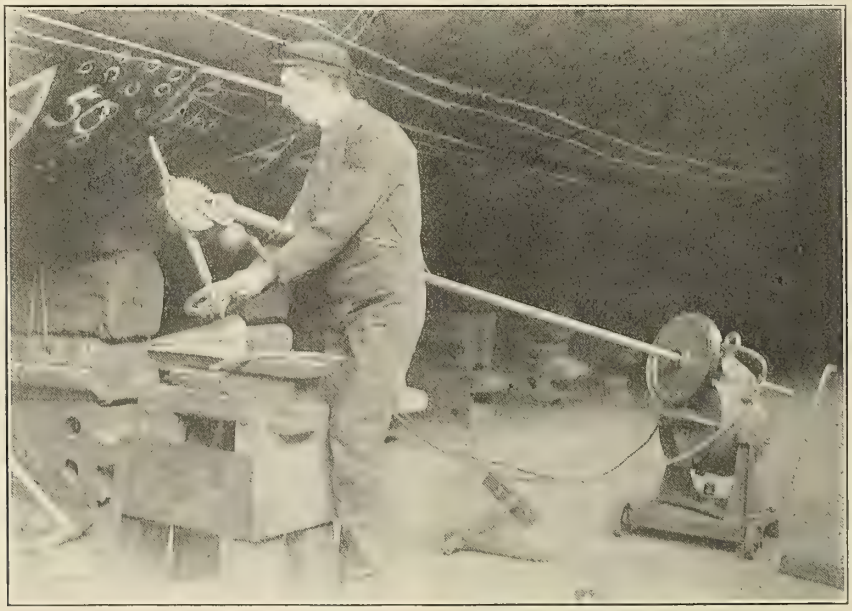


FIG. 2.—ENGLISH PORTABLE ELECTRIC DRILL.

saving devices. An electric deck planer constructed by Messrs. Mavor and Coulson, of Glasgow, is shown in the accompanying illustration, Fig. 5. Until recently these machines were fitted with direct-current motors only, but the introduction of three-phase current from supply stations has created a demand for deck

FIG. 4.—CLEVELAND PNEUMATIC RIVETERS IN OPERATION AT GLASGOW, SCOTLAND.

planers fitted with alternating-current motors. The machine illustrated, which is equipped with a 3-horsepower motor running at 3,000 revolutions per minute, and driving the cutting knives at the same speed, will plane under working conditions on board ship an area of 360 square feet of deck in one hour. The attendance required is one man at 9 pence per hour to guide the machine, one apprentice at 1 1-2 pence per hour to draw it, and one apprentice at 1 1-2 pence per hour to sweep away the shavings, a total of one shilling per hour. By hand labor an area of 45 square feet of deck planed is a fair day's work for a carpenter, for which 10 shillings are paid. The machine therefore does in one hour, at the labor cost of one shilling, as much work as eight men do in a day, at a cost of four pounds, and the machine leaves a much better surface. On a vessel of moderate size the saving effected by the electric machine amounts to about £80.

The portable electric drilling machine of English construction, shown in Fig. 2, has no flexible or telescopic shafts, knuckle joints, or similar devices for transmitting the power from the motor to the drill, the drive being practically direct. The drill may be stopped or started instantly without removing the hand from the drill head, and the electric motor which is mounted on a truck, as well as the drill head and the whole portable tool, may be managed by one man without the slightest difficulty. The truck is two-wheeled, and so arranged that when the handles are released two short legs hold it in place on the ground or floor, as with a wheelbarrow, so that it cannot be moved about when in operation, as a four-wheeled truck might be. The motor rests on trunnions when in its carriage. A bracket carrying a hollow shaft is situated on top of the motor, and on one end of this shaft there is a spur wheel which is driven from a pinion on the armature shaft. A long internal shaft whose end is connected with the drill head slides inside the hollow shaft. On the inside of the hollow shaft there is a key which engages in a slot which runs nearly the whole length of the long internal shaft. From the motor the motion is transmitted through the hollow shaft to the long sliding shaft, and this can be carried to the work to the limit of its length, or can be drawn back so that the drill head is very close to the motor.

There are two pairs of bevel gear wheels, or four wheels in all, constituting the drill head, the hand wheel at the top being used to feed the drill spindle down when it is held in the clamp. The drill may, with this arrangement, be turned through a complete

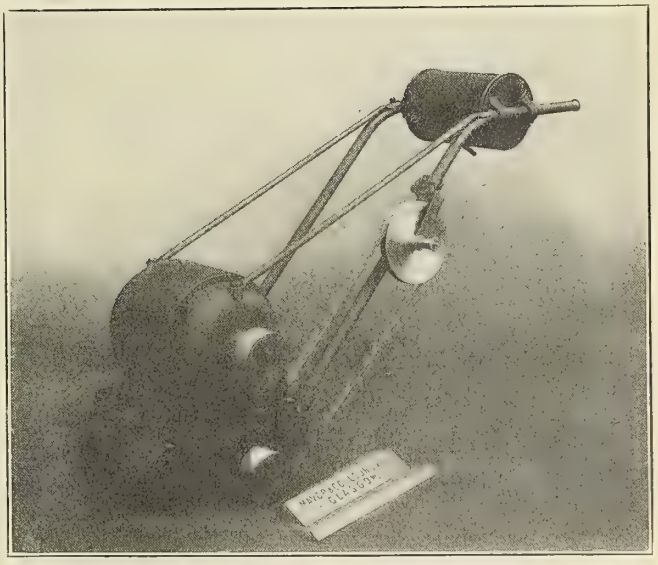


FIG. 5.—ENGLISH ELECTRIC DECK PLANER.

circle in the plane at right angles to the long shaft, also through almost a complete circle in the same plane as the shaft. When tapping has to be done and a slower and more positive motion is desired, a universal worm-gear drill head can be fitted to the outfit in place of the bevel gearing. This electric portable drill has a

capacity for drilling holes in steel up to 1 1-2 inches in diameter, but larger machines are constructed to drill holes up to 2 1-2 or 3 inches in diameter, a 3-horsepower motor being required for the latter type, and a 1 1-2-horsepower motor for the smaller machine. The carriage weighs 138 pounds, the drill head 44 pounds, and the shaft 24 pounds, while the weight of the electric motor is 240 pounds. As the switch is placed near the drill head, the operator may start and stop the motor instantly when desired.

Flexible shafting is also extensively employed for transmitting power from portable electric motors to drill heads and other tools, particularly in America, while in Germany telescopic shafts are largely employed, with special portable tool heads designed for various classes of work in machine-shop practice.

SOME RECENT EXPERIMENTS AT THE U. S. MODEL BASIN.*

BY D. W. TAYLOR, NAVAL CONSTRUCTOR, U.S.N.

The experiments which are the subject of this paper have been under contemplation at the experimental model basin for some time, the design, construction, and testing of the necessary apparatus and appliances having necessarily extended over a considerable length of time. The experiments themselves are quite recent, the first regular test of a model propeller having been made in August last.

The primary object of the experiments was the determination of the power and efficiency of model propellers of the ordinary three-bladed type throughout the range of pitch ratio and blade

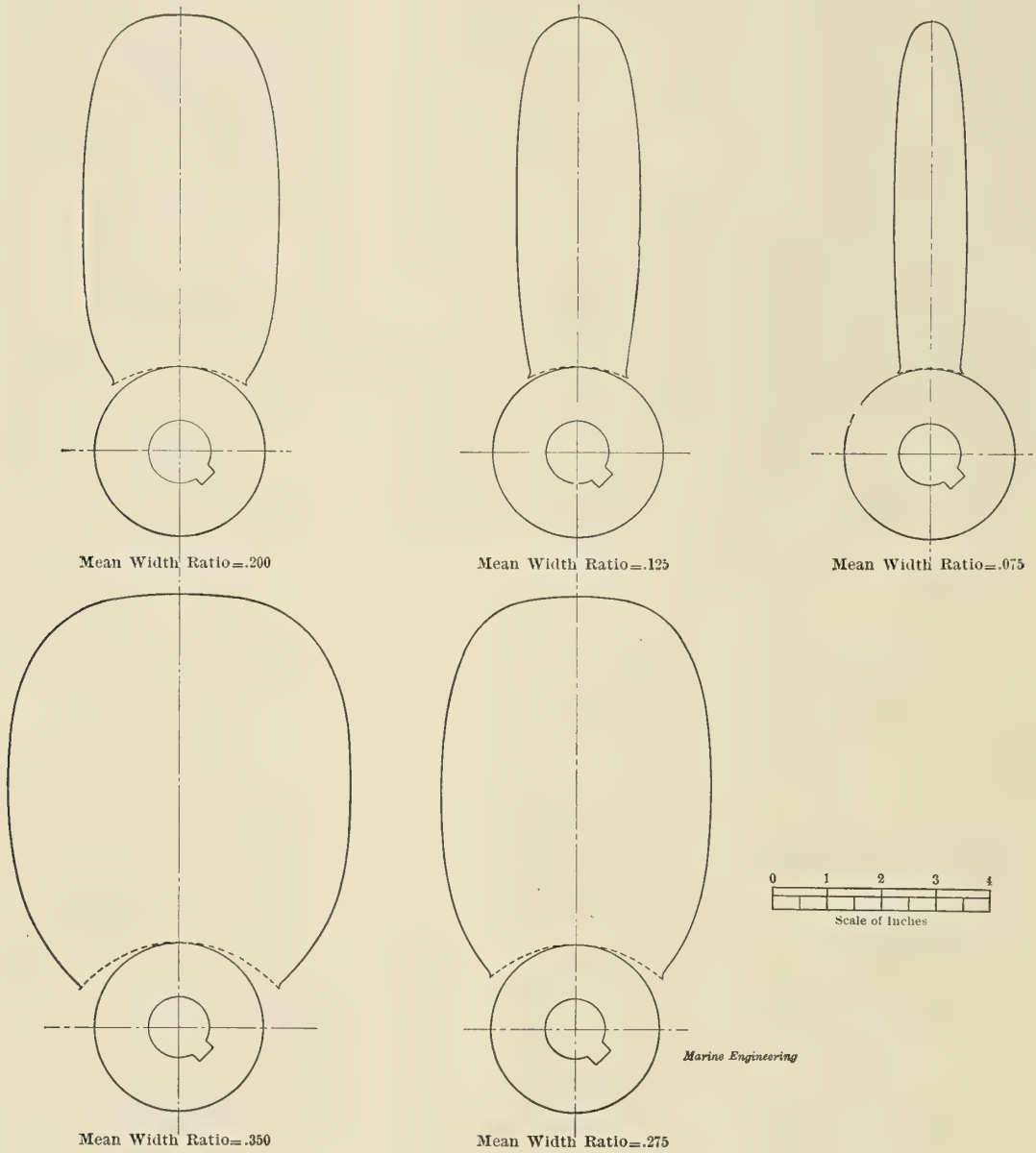


FIG. 1.

"Engineers of America," a directory and biography, will shortly be published by E. C. Brown, editor of the *Progressive Age*, 280 Broadway, New York city. Mr. Brown states that he believes this work will fill a long-felt want, and he intends to supplement the personal data with a complete list of all the engineers associated with the various national engineering societies. Correspondence is desired with all practicing engineers, that the work may be as complete as possible.

area likely to be encountered in practice. In view, however, of the present tendency toward increasing speed of revolution of propellers, the experiments were extended to the unusually low pitch ratio of .4.

The series of experiments covered 30 model propellers. They were all 16 inches in diameter, three-bladed, of uniform pitch,

*Read at the meeting of the Society of Naval Architects and Marine Engineers, November, 1904.

diameter of hub 3 1-8 inches, thickness of center of blade at the root 9-32-inch, at the tip 3-32-inch. The pitches used were six in number, namely, 6.4, 9.6, 12.8, 16, 19.2, and 24 inches. For each pitch five widths of blade were used, the mean width ratios, actual total blade area and blade area expressed as a fraction of the disc area, being as indicated in the table below. Fig. 1 shows the blade area outlines used.

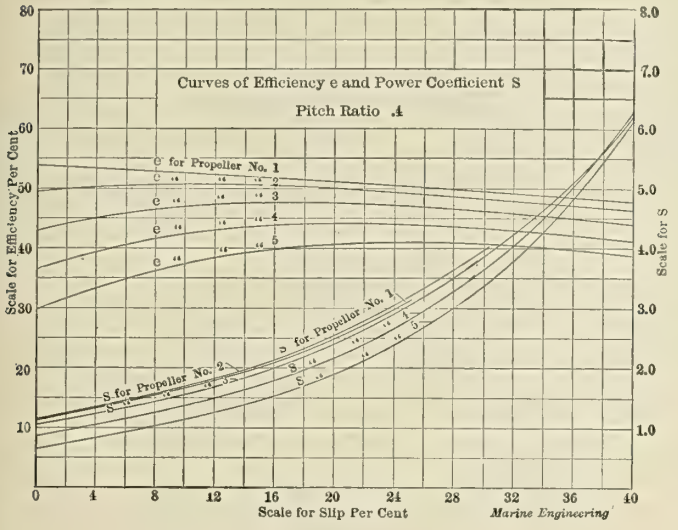

Curves of Efficiency e and Power Coefficient S
Pitch Ratio .4
e for Propeller No. 1
S for Propeller No. 1
S for Propeller No. 2
Scale for Efficiency Per Cent
Scale for S
Scale for Slip Per Cent
Marine Engineering

FIG. 2.

The propellers were of composition and were cast at first from wooden patterns, but afterwards it was found simpler to use plaster of Paris patterns. They were finished as to the face in a special machine described in Appendix "A."† The backs of the blades were finished by hand, and both backs and faces were carefully smoothed with emery cloth before the tests. The dynamometer apparatus and methods used for making tests are

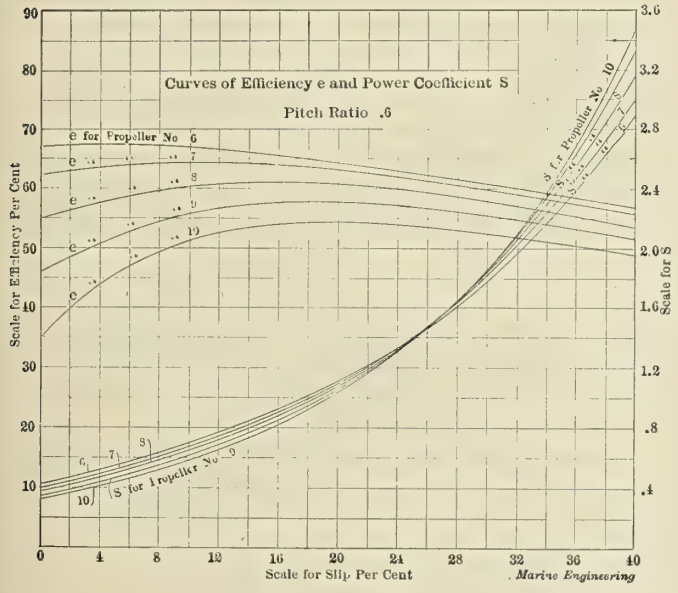

Curves of Efficiency e and Power Coefficient S
Pitch Ratio .6
e for Propeller No 6
S for Propeller No. 10
S for Propeller No. 9
Scale for Efficiency Per Cent
Scale for S
Scale for Slip Per Cent
Marine Engineering

FIG. 3.

described in Appendix "B." The methods of reduction of the tests are described in Appendix "C." The results were carefully faired by cross-curves and propellers whose curves did not fair in satisfactorily were carefully retested.

I now invite your attention to the final results of the experiments. Figs. 2 to 7 show curves of efficiency and of a coefficient,

†This and other appendices of the paper are not here published owing to lack of space.

TABLE OF PROPELLER VARIABLES.

Propeller number.	Pitch Ratio. .4	.6	.8	1.0	1.2	1.5	Mean Width Ratio.	Total Developed Area Sq. Inches.	Developed Area ÷ Disc Area.
	1	6	11	16	21	26	.075	24.28	.1207
	2	7	12	17	22	27	.125	40.46	.2012
	3	8	13	18	23	28	.200	64.74	.3220
	4	9	14	19	24	29	.275	89.02	.4427
	5	10	15	20	25	30	.350	113.30	.5635

which I designate by S, conveniently called the power coefficient, for each of the thirty propellers. As explained in Appendix "C," if P denote the horsepower absorbed by the propeller, d the diameter of propeller in feet, and V the speed of advance of the propeller in knots, we have

$$P = .0093648\ Sd^2\ V^3.$$

Evidently then curves of e, or efficiency, and of S, completely characterize a given model propeller. The above formula will apply to a propeller of any size similar to the model propeller, provided the well known law of comparison connects the operations of the small and large propellers. We must look upon the

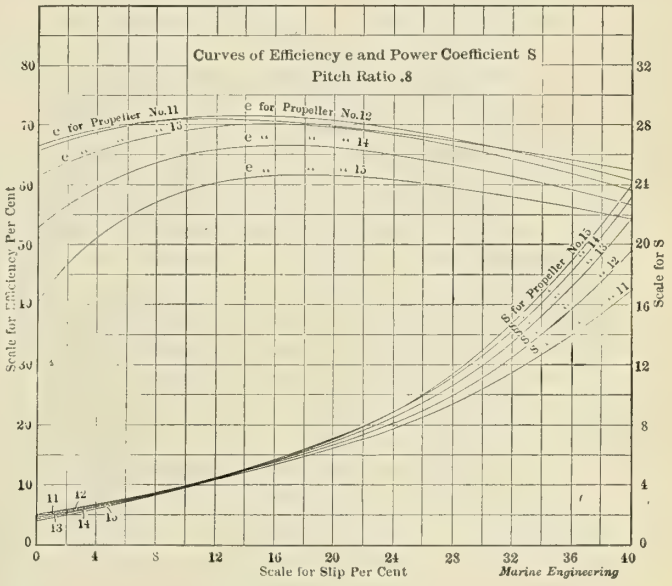

Curves of Efficiency e and Power Coefficient S
Pitch Ratio .8
e for Propeller No.11
e for Propeller No.12
S for Propeller No.15
Scale for Efficiency Per Cent
Scale for S
Scale for Slip Per Cent
Marine Engineering

FIG. 4.

application of the law of comparison to propellers with some suspicion until it is practically demonstrated by comparison between model propeller experiments and full-sized propellers, but in the present condition of the science it is our most reliable guide.

The 16-inch propellers experimented with are materially larger than any previous model propellers concerning which I have knowledge. Froude's model propellers, experimented with about twenty years ago, were 8.16 inches in diameter. Prof. Durand's experiments, described in volume V of our Transactions (1897), were made with propellers 12 inches in diameter. A 16-inch propeller tested as these were is by no means a toy. Each propeller was tested up to a thrust of 150 pounds or a slip of 40 percent. The propellers of the coarser pitch did not show 150 pounds thrust with 40 percent slip, but they did show more than 100 pounds at the highest speed experimented with. The finer pitched propellers reached 150 pounds thrust at slips materially less than 40 percent for the higher speeds of advance.

After the runs the propellers were carefully tested on the machine described in Appendix "A" for variation of pitch. In no case was there any appreciable distortion of pitch. The five propellers of 1 pitch ratio were also tested for pitch in the machine with a weight of 50 pounds hung to each blade from its center at

a radius of 2-3 the maximum radius of the blade. Some of the narrow blades bent a good deal under this test, but did not change pitch appreciably at any point and took no permanent set.

In connection with Figs 2 to 7 it may be well to point out the fundamental elements affecting the powers and efficiencies of propellers. They are the diameter, the pitch, the blade area and

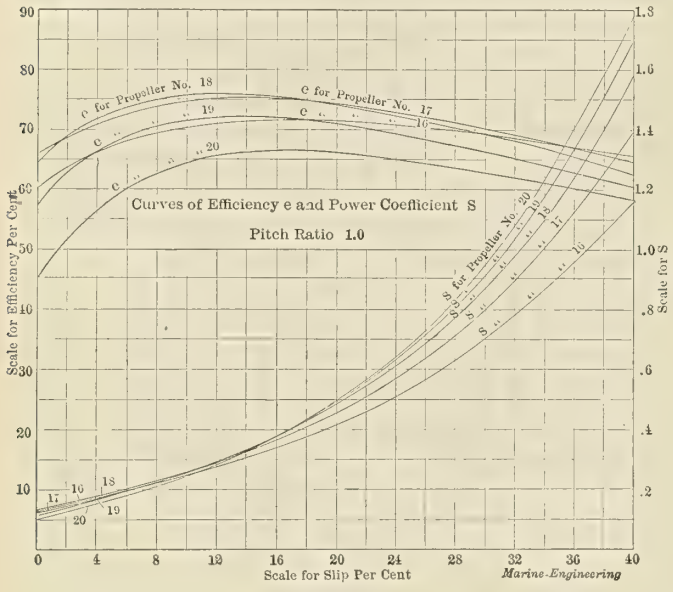

Curves of Efficiency e and Power Coefficient S
Pitch Ratio 1.0
e for Propeller No. 18
e for Propeller No. 17
Scale for Efficiency Per Cent
Scale for S
Scale for Slip Per Cent
Marine Engineering

FIG. 5.

the shape of blade section. The latter is seldom referred to or included as a fundamental feature in propeller design, but these experiments indicate clearly that it is a thing which must be reckoned with, and that probably many unsolved puzzles in connection with propellers have been dependent upon shape of the blade section. The conventional meaning of the word "pitch," as applied to a propeller of uniform pitch, is the pitch of the driv-

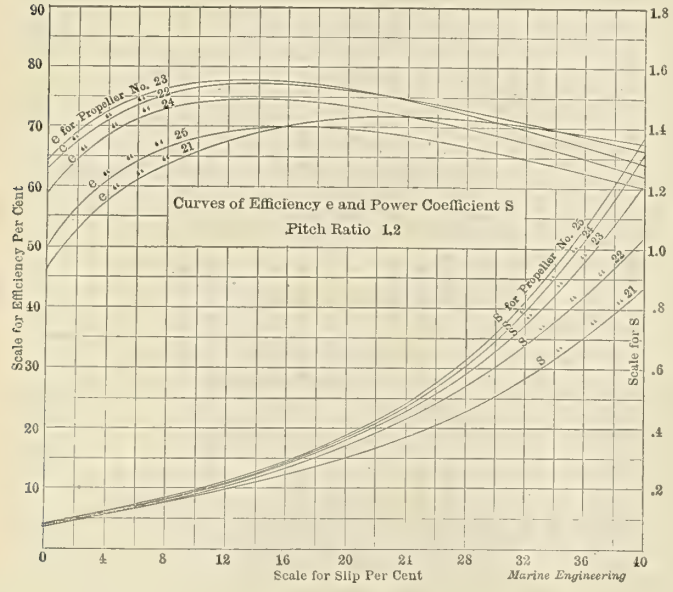

Curves of Efficiency e and Power Coefficient S
Pitch Ratio 1.2
e for Propeller No. 23
Scale for Efficiency Per Cent
Scale for S
Scale for Slip Per Cent
Marine Engineering

FIG. 6.

ing face or helicoidal face. The back of the blade, however, also has a pitch at each point, and in the conventional section it varies from point to point. If the water always remains in contact with the back of the blade as well as the face of the blade, it would seem obvious that the pitch of the back of the blade necessarily affects the operation of the propeller. It is not generally realized how widely the pitch of the back of the blade varies from the face. Fig. 8 shows for one series of the model propellers experimented with, namely, that which had a pitch ratio of unity, the variation of pitch of the back of the blade at a section taken at two-thirds of the radius from the center. For the finer pitch screws of the same section these variations are more pronounced; for the coarser pitches they are not so pronounced. It is seen that for propellers of .200 mean width ratio and a uniform pitch on the face of 16 inches, the pitch of the back varies from 7.8 inches to 26.4 inches. Conventionally we call the pitch of this propeller 16 inches, but it is obvious that it is practically impossible to know what the real pitch is.

While it is the primary object of this paper to lay before the Society actual results of comprehensive experiments which each member may analyze for himself, rather than to set forth theories of my own, I wish to direct attention to a few features of these results which appear worthy of remark and requiring explanation. In the first place we have been accustomed for many years to suppose that the maximum efficiency of the propeller varied but little from 70 percent. These experiments show a maximum efficiency of propeller No. 28 of 78.8 percent. While, as described in Appendix "B," the efficiencies do not take account of the hub action, and are hence slightly higher throughout than

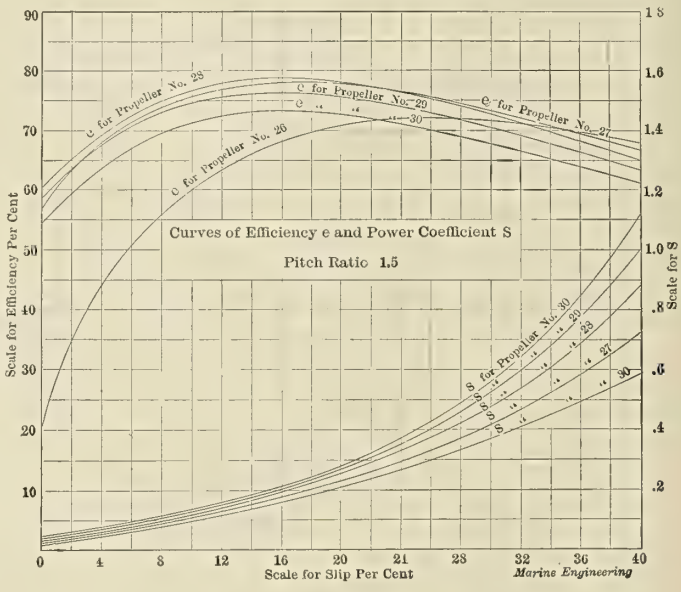

Curves of Efficiency e and Power Coefficient S
Pitch Ratio 1.5
e for Propeller No. 28
e for Propeller No. 29
e for Propeller No. 27
e for Propeller No. 26
Scale for Efficiency Per Cent
Scale for S
Scale for Slip Per Cent
Marine Engineering

FIG. 7.

if the hub action had been considered, I am confident that the high efficiencies were actually obtained. It will be noted that these model propellers are most accurately finished and the blades are comparatively thin, the thickness at the root being .018 and at the tip .006 of the diameter. There is a steady falling off in efficiency as the pitch ratio is decreased. This was to be expected from theoretical considerations.

Perhaps the most remarkable feature of these results is the disclosure of the fact that as the pitch ratio is decreased the influence of the area of blade upon the results becomes progressively less until for the propellers of very fine pitch ratio the narrow blades actually absorb the greater power. For the propellers of .6 pitch ratio, curiously enough, at a slip of 26 percent it makes practically no difference what the width of blade is, the same power is absorbed by the propeller, although the narrow blades show somewhat greater efficiency. Below this slip the narrower blades absorb the greater power, just as for the .4 pitch ratio, while above this slip the wider blades take more power. As pitch ratio is increased this action becomes less and less, until for the higher pitch ratios experimented with the power absorbed is greater the wider the blade. The increase in the power with increasing width of blade is, even for the high pitches, however, much less than it should be according to propeller theories of

which I have any knowledge. Fig. 9 shows for 25 percent slip for the series of 1.5 pitch ratio a curve of power absorbed plotted over mean width ratio.

Another remarkable feature is the fact that as the pitch ratios of the propeller decrease, the power absorbed and efficiency at low slips are both very great, the slip being based upon a conventional pitch.

ratio were tested up to 6 knots only. Fig. 10 shows curves of thrust and torque plotted upon slip for propeller No. 1 for 5, 6 and 7 knots speed of advance. The 5-knot curves are normal. The 6-knot curves show evidences of cavitation at about 0 percent slip and 115 pounds thrust. The 7-knot curves show strong evidences of cavitation at about—15 percent slip and 80 pounds thrust.

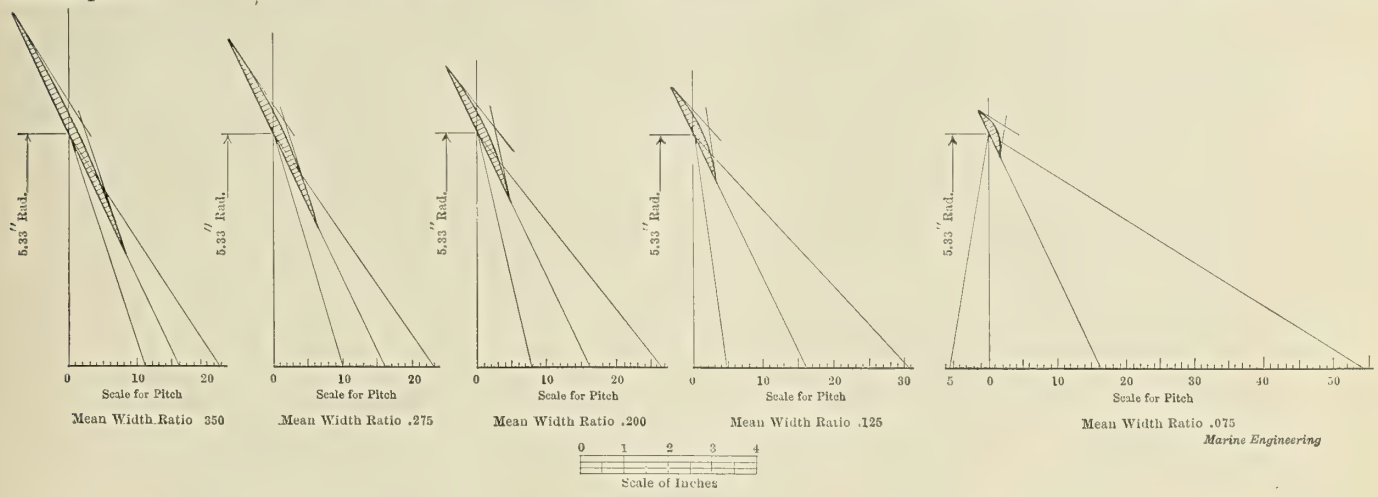


FIG. 8.

Thus considering the series of propellers from Nos. 6 to 10 it is seen that for the narrowest blade the efficiency at zero slip is 66.8 percent while the maximum efficiency of this propeller is but 67.4 percent. As the blade width is increased this effect is decreased. Of course an explanation of many of the features above described is the fact already pointed out that the slip based upon conventional pitch is necessarily erroneous. It would seem that the real or effective pitch of the propeller was a quantity which varied very much. For small slips the effective pitch would seem to be very much greater than the nominal pitch, whereas for large slips it would seem to become less again. In other words, at small slips the propeller acts as if the following portion of the

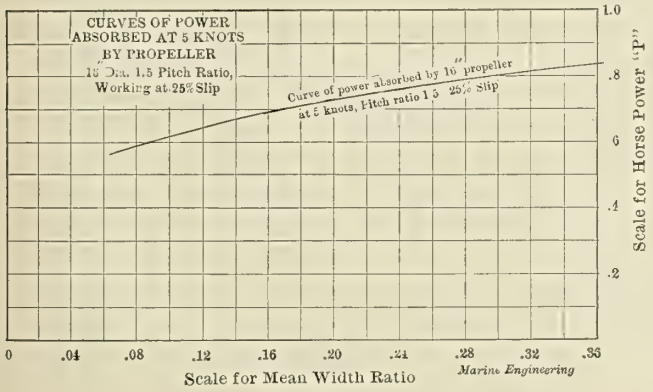

CURVES OF POWER ABSORBED AT 5 KNOTS BY PROPELLER 16" Dia. 1.5 Pitch Ratio, Working at 25% Slip
Curve of power absorbed by 16" propeller at 5 knots, Pitch ratio 1.5 25% Slip
Scale for Horse Power "P"
Scale for Mean Width Ratio
Marine Engineering

FIG. 9.

back of the blade was predominant, whereas at large slips it acts as if the leading portion of the back of the blade assumes predominance.

CAVITATION.

It is generally recognized that model propeller experiments are of little value as regards cavitation, owing to the fact that model propellers working under the combined pressure of the air and water have a virtual submergence very much greater than that of full-sized propellers. For many of the propellers experimented with there seemed to be a tendency toward reduction of thrust and torque when tested at the 7-knot speed, but there was no pronounced cavitation observed, except in the case of No. 1 propeller when tested at 7 knots. Most of the propellers of this pitch

It will be noted that cavitation occurred at a very low thrust per square inch of projected area, about 4.3 pounds for the 7-knot speed, and that the thrust at which cavitation became marked was about 40 percent greater at 6 knots than at 7 knots. I feel very confident that in the case of this particular propeller the breaking down, which we call cavitation, while affected by the thrust, was

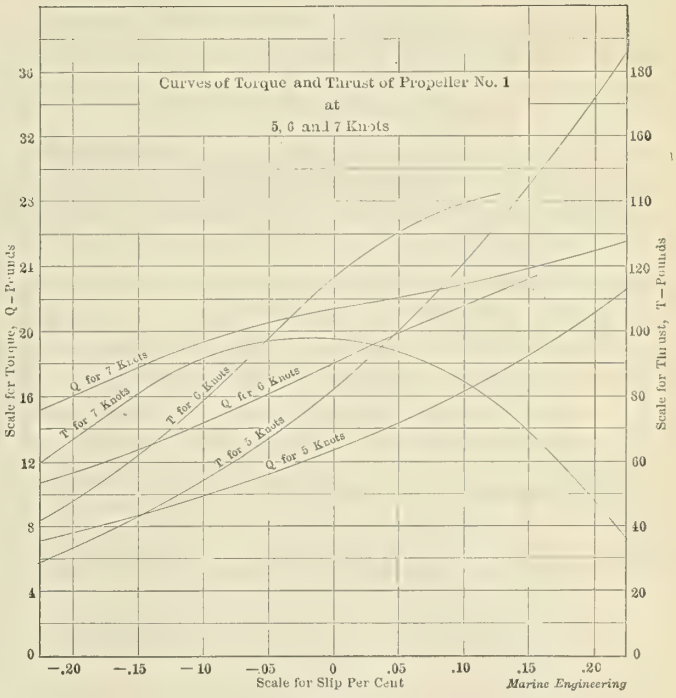

Curves of Torque and Thrust of Propeller No. 1 at 5, 6 and 7 Knots
Q for 7 Knots
T for 7 Knots
T for 6 Knots
Q for 6 Knots
T for 5 Knots
Q for 5 Knots
Scale for Torque, Q—Pounds
Scale for Thrust, T—Pounds
Scale for Slip Per Cent
Marine Engineering

FIG. 10.

strongly affected by two other factors, namely, the speed of revolution and the shape of the blade section. In other words, in this particular case the breaking down of the propeller was a case of cleaving the water, as when one draws a stick through water, and at a sufficiently high speed of revolution the propeller would have shown cavitation with almost no thrust. It seems reasonable to suppose that all cavitation is largely of the same nature, and a function not only of the thrust but of the speed of revolution and

shape of blade section, and that it could probably be mitigated in many cases, or deferred by modifying suitably the blade section.

As bearing upon the effect of shape of blade section, I submit Figs. 11 and 12. These refer to three propellers, Nos. *23*, *23a*, and *23b*. No. *23* was one of the regular series, 16 inches diameter, 19.2 inches pitch, 64.7 square-inch blade, and .200 mean width ratio. Nos. *23a* and *23b* had the same diameter and blade area, but had sections as shown in Fig. 11. The pitch of the lines AA for *23a* was 19.2 inches. In other words, the face and back of the blade were similarly convex. For *23b* the pitch of the portion *a-b* of the face of the blades and the portion *c-d* of the back of the blade was the same, namely, 19.2 inches. Fig. 12 shows together the curves of efficiency and *S* for these three propellers. It is seen that for propeller *23a* there is a slight gain of efficiency as compared with No. *23*, the usual shape, but a marked falling off in the power utilized. At the slip corresponding to maximum efficiency *23a* can utilize only about 50 percent as much power as No. *23*. Propeller *23b* shows a falling off in efficiency as compared with *23* and *23a*, except at very low slips, but has somewhat more capacity for utilizing power than No. *23*. Propeller No. *23* was also run reversed, with the backs of the blades driving. The results are not shown on the curves, but the maximum efficiency shown was not as great as 50 percent.

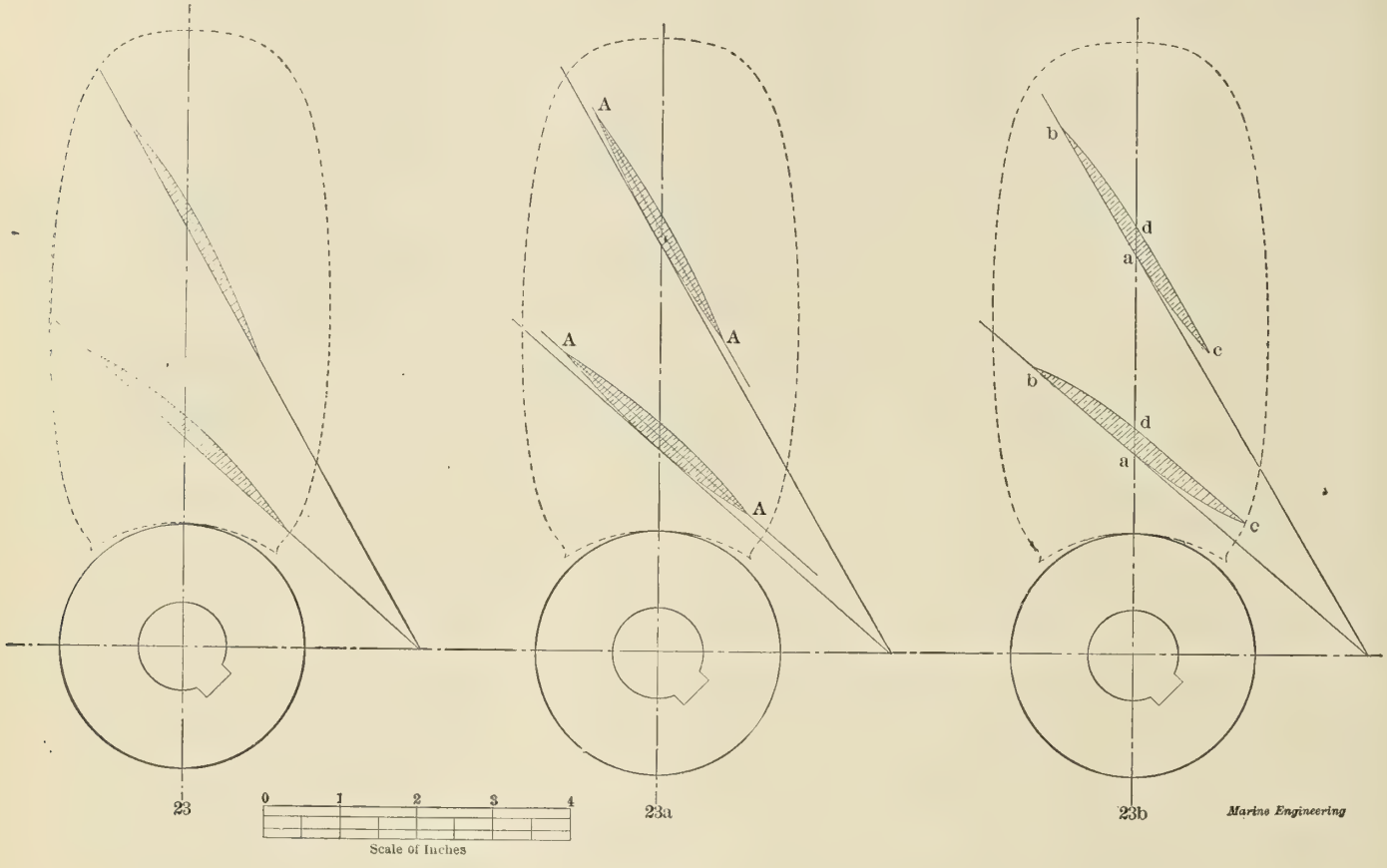


FIG. 11.

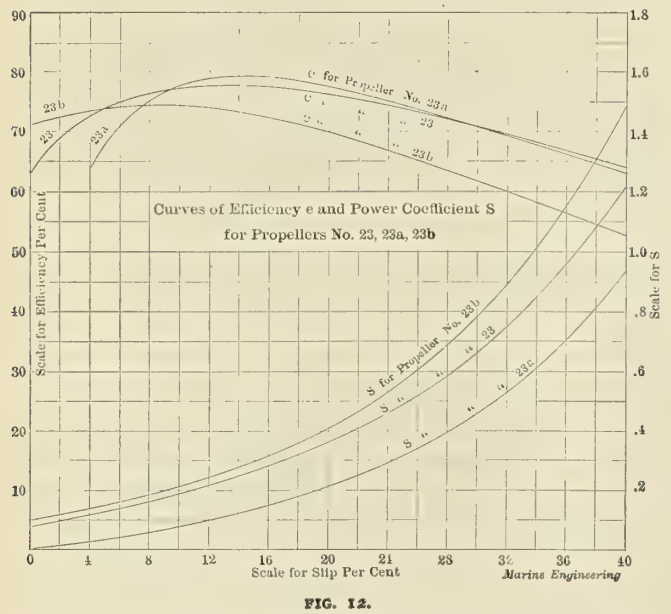
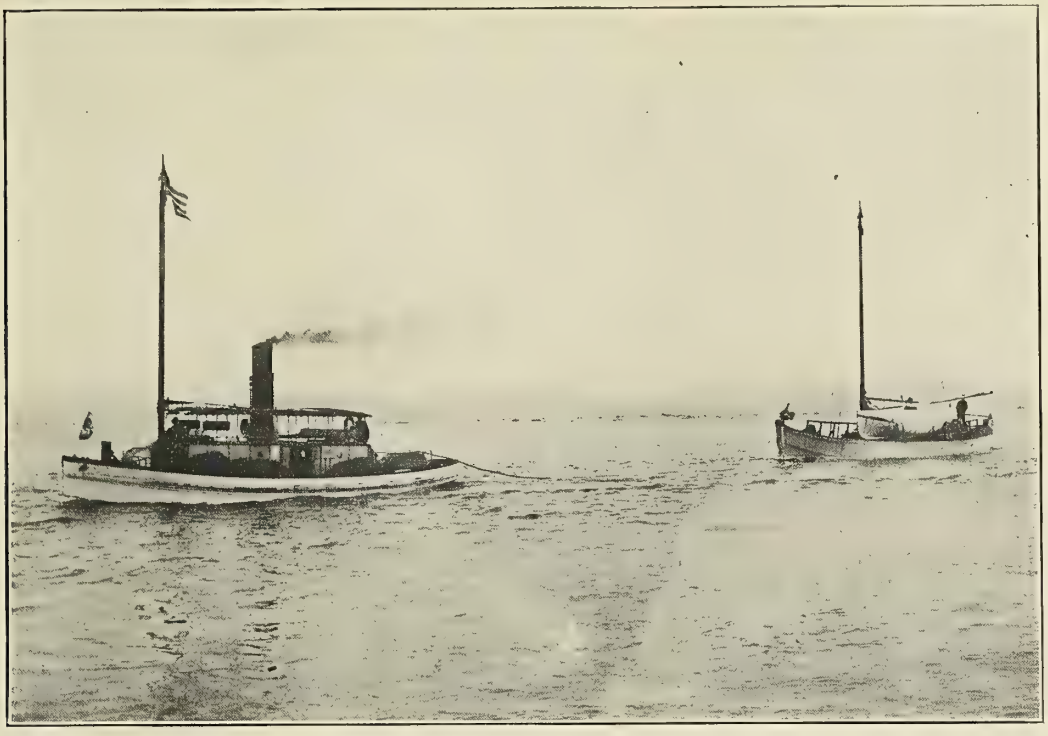

Curves of Efficiency e and Power Coefficient S
for Propellers No. 23, 23a, 23b
Scale for Efficiency Per Cent
Scale for S
Scale for Slip Per Cent
Marine Engineering

FIG. 12.

Fig. 12 illustrates strongly what is evident from careful examination of Figs. 2 to 7, namely, that the shape of blade section is a very important factor in propeller performance. It should be noted in this connection that fine pitched propellers should have as high a value of *S* as practicable in order to keep down the diameter and increase the pitch ratio.

While a number of the model basin staff have taken part in the propeller experimental work, the reduction of the experimental results has been done almost entirely by or under the direction of Mr. A. V. Curtis and Mr. L. F. Hewins. These gentlemen have given me much valuable assistance in preparing the appendices to this paper, and without their assistance it would have been almost impossible for me to have prepared it in time for the meeting of the Society.

The steamer *Eastland*, mention of the high speed of which was made on page 42 of our January issue, has been fitted with the Ellis & Eaves system of induced draft. We are informed that prior to the installation of this system the boat was able to make only about 18 miles per hour, and it was not until after this system was in operation that the designed speed of 21 miles was reached.



THE TUG CHAMPERICO.

Tug Boat Champerico.

The accompanying engraving was taken of the wooden tug *Champerico,* as she started on her long voyage from San Francisco to Champerico, Central America, which is to be her home port. She was built by John Twigg and Sons' Company, the well-known ship and boat builders of San Francisco. The hull is constructed of pine, while the frames, stem, stern post, keel, and rudder are of oak. The general dimensions of hull are as follows: length over all 55 feet; molded beam 14 feet; depth of hold 8 feet, with a draft of 6 feet 6 inches loaded. She is strongly constructed throughout to meet the requirements of a boat of her class, which is subjected to severe usage in the handling of lighters and barges used for loading and unloading vessels in Central American ports.

The engine is of the fore-and-aft compound type, with cylinders of the following dimensions: high-pressure cylinder, 10 inches diameter; low-pressure cylinder, 22 inches diameter, with a common stroke of 14 inches. The air, circulating, bilge, and feed pumps are driven from the low-pressure crosshead by means of a beam. They were formerly used in the tug *Adolpha,* which was wrecked in Central America about one year ago, and are of the best workmanship and in first-class condition. Steam is supplied by one Scotch boiler, of the marine type, 90 inches diameter by 120 inches in length, with a working pressure of 160 pounds per square inch. An average speed of 9 miles an hour was maintained during the trial trip, which was of 4 hours' duration. As shown in the engraving she has in tow a 45-foot lighter loaded with 30 tons of coal, with which, with 12 tons in her own bunkers, and a deck load of 8 tons more, she will make the trip of about 2,400 miles without calling at any port *en route* for fuel.

Captain North, of New York, with four men, make up the little crew that has undertaken the task to deliver her to her owners. Mariners are somewhat skeptical as to the safety of making such a long ocean voyage in a vessel of the *Champerico's* dimensions, but it has been successfully done several times before in vessels of even smaller size built by the same builders. Captain North states that he expects to make an average speed of 7 1-2 miles per hour during the trip if he meets with moderate weather and has no serious mishaps.

An Alternate Bid on the Armored Cruisers.

Mr. Francis T. Bowles, president of the Fore River Shipbuilding Company, and formerly chief constructor of the United States navy, submitted an alternate and most interesting bid for the construction of the two armored cruisers *North Carolina* and *Montana,* for which contracts were recently let to the Newport News Shipbuilding and Dry Dock Company. His bid was as follows:

"This letter is submitted in explanation and description of our bid of this date on an armored cruiser, or cruisers, of 14,500 tons displacement each, and is accompanied by a set of plans numbered from 1 to 11, inclusive, showing the modifications in the department's plan upon which this bid is based. This bid is offered under the terms of the act appropriating for these vessels and with particular reference to the following words: 'And the contract for the construction of these vessels shall be awarded by the Secretary of the Navy to the lowest responsible bidder, having in view the best results.'

"Believing, in 1902, when the designs for the armored cruisers *Tennessee* and *Washington* were prepared, that these vessels were not only the most effective of their class designed up to that date, but that they also met with care and sufficiency such criticisms as had been made by officers of the navy upon vessels previously built and designed, with special reference to the isolation and the operating of the elements of the battery, the safe and rapid supply of ammunition, and armor protection, it is also apparent that since the date of these designs and due to the experience of other countries with large armored cruisers, a legitimate criticism upon these designs has been made that a considerable portion of the battery is located too near the water line for effective use at sea. Therefore, with a view of overcoming this objection, and also with a view of obtaining a further and better protection for the vitals of the vessel by armor, a better isolation and efficiency of the battery and a greater end-on fire, we have prepared and submit these designs, believing that the fighting value of the vessel will be greatly enhanced.

"We have also arranged to present to the department in this design the Curtis marine turbine, an American invention already highly developed for stationary use, in place of triple-expansion reciprocating engines. There will be one low-speed reversible

turbine on each of the twin-screw shafts. After investigation we are satisfied that this turbine has great superiority over the old-style engine in features of particularly essential value in naval vessels of the cruiser class. In this vessel it gives assurance of higher speed by its ability to convert into power all the steam the boilers can produce, whereas the simplicity of its construction and small number of parts are an assurance of its far greater readiness for service under all conditions.

"With particular reference to the circular defining the chief characteristics of these vessels in accordance with the department's designs, the modifications which we submit are, in brief, as follows:

THE BATTERY.

"1. Referring particularly to sketch plan No. 1, it will be observed that eight of the 6-inch guns carried on the gun-deck of the department's design at a height of 14 feet 6 inches above load water line, have been raised to the main-deck level, and together with four of the main-deck guns, are mounted in pairs in six turrets. Therefore, the increase in height of the eight 6-inch guns will be from 14 feet 6 inches to 23 feet above the water line. The only 6-inch guns remaining on the gun-deck are those forward and aft, having a fore-and-aft fire. The result of this change is to give two additional 6-inch guns firing ahead and two firing astern, making the total ahead or astern fire two 10-inch and six 6-inch guns. The 6-inch guns will be inclosed in turrets of which the general features are shown in plan No. 2, consisting of 6-inch armor with 6-inch barbette and a 4-inch ammunition tube leading to the protective deck, the turning mechanism and ammunition supply being operated from a passage below the protective deck.

"2. The design provides for carrying the same number of guns of all calibers as the department's plans, and gives equal protection to all the minor guns, excepting only four 3-inch guns on the upper deck amidships, two on each side.

THE ARMOR PROTECTION.

"3. The armor protection provided on the alternate design is shown in detail on plan No. 3, and the comparative armor protection of this design and the department's plan is clearly indicated on plan No. 1. The principal feature of the change consists in increasing the thickness of the main belt throughout its length between the 10-inch turrets from 5-inch to 6-inch, which at 3,000 yards would render the belt impenetrable by a 6-inch gun, whereas on the department's design the main belt is penetrable by a 6-inch gun and proof only against a 5-inch gun. All the other features of the armor protection correspond with those of the department's design, including the protective deck.

PROPELLING MACHINERY.

"4. In place of the department's design of vertical twin-screw, four-cylinder, triple-expansion engines of a combined indicated horse power of not less than 23,000, we propose to substitute twin-screw turbines of 11,500 horse power each, having 12 feet diameter buckets, and making at maximum power about 300 revolutions per minute. We shall be able to satisfy the department that the steam economy of the proposed turbine will compare satisfactorily with that of the department's design for reciprocating engines. The boiler installation will correspond precisely with the department's conditions. All miscellaneous engineering requirements, auxiliary steam machinery and equipment requirements, will be complied with in accordance with the department's circular, plans and specifications."

The United States Civil Service Commission announces an examination on February 15-16, 1905, at the places mentioned in an accompanying list, to secure eligibles from which to make certification to fill four vacancies at $1,200 per annum each and two vacancies at $1,400 per annum each, in the position of civil engineer, Philippine service, and vacancies as they may occur in that service requiring similar qualifications.

Report of Bureau of Construction and Repair.

One of the most interesting features to our readers of the recently issued annual report of the chief of the Bureau of Construction and Repair of the Navy Department, is the estimate as to the dates of completion of the new ships now under construction. Altogether it is expected that 21 ships of various types will be finished and placed in service during the present calendar year. Of these, five are the first-class battleships of the *Georgia* type, which are the only five ships building with superposed turrets for the heavy guns. These ships were authorized in 1899 and 1900, and it is expected that they will be completed about the end of the year.

The next ships in point of importance are four of the armored cruisers of the *Pennsylvania* class, three of which are scheduled to be finished before the first of April. It is expected that the last cruiser of the *Denver* class, the *Galveston,* will be finished about April 1, and that the first one of the *Charleston* class will be added to the navy list about the first of July.

In addition to these, the two gun boats building by the Gas Engine and Power Company, the two training ships building at the navy yards at Boston and Mare island respectively, and the training brig building at the navy yard at Portsmouth, will be finished by the end of the fiscal year. The five torpedo boats and destroyers remaining uncompleted at the present time will, it is expected, be finished early this year. It will be remembered that these five boats were all authorized by Congress at the opening of our war with Spain, and that, therefore, they have been seven years in building.

The report shows that a total of 8,405 men are employed in the various navy yards and naval stations, of which number about 2,600 find employment in the New York navy yard, and more than 1,400 at Norfolk. It may surprise some to learn that the number employed at Cavite was about 730, this being larger than the number employed at any of the continental yards, except the two above mentioned and Boston.

A Few Collisions.

We are publishing this month photographs of four ships which have been badly damaged in or near the English channel, and have put into Falmouth for repairs. These repairs have been carried on by the local firm of Cox & Company, and by the Falmouth Docks Company, the latter concern having sufficiently large graving docks to receive almost any ship afloat.

The steamship *Marpessa,* of 1,730 tons, ran into a Dutch army transport carrying troops to the East Indies. The collision occurred in the Bay of Biscay. The troops were all saved by the crew of the *Marpessa* and carried to Falmouth where a new bow had to be fitted, the condition of the bow being well shown in the photograph. The Dutch transport was sunk.

The steamship *Universal,* of 1,274 tons, was abandoned off Ushant and towed into Falmouth by the steamship *Emily.* The *Universal* had been struck in a fog by the steamship *Cambodge,* and had had her stern completely sheared off, the after bulkhead alone preventing the vessel from sinking. In spite of the shock a large mirror fastened to this bulkhead was uninjured, and may be seen in the photograph. The crew of the *Universal* were all taken off by the *Cambodge,* but went back to their ship after the *Emily* had taken her in tow and it was seen that she was in no danger of sinking.

The bark *Abernyte* ran into the broadside of the steamship *Tripoli,* but did not sink her, both vessels reaching Falmouth, and both were repaired at the same time in the dry dock.

The fourth photograph is of the Italian steamship *Costanza,* of 2,903 tons, which ran into the rocks off the Cornish coast and was badly damaged.

The photographs were taken by Mr. W. M. Harrison, of Falmouth.

STEAMSHIP UNIVERSAL.
STEAMSHIP MARPESSA.

STEAMSHIP COSTANZA.
BARK ABERNYTE.

NAVAL ARCHITECTURE.*

BY SIR WILLIAM H. WHITE, K.C.B., F.R.S.

PART IV.

INTERNAL COMBUSTION ENGINES.

In common with France and the United States, Great Britain is rapidly developing the construction of oil-motor launches of great speed, and the possible application of internal combustion engines, on a larger scale, to ship propulsion, is receiving attention. Messrs. Thornycroft have designed machinery of this class for a torpedo-boat destroyer of 6,000 H. P., the piston speed being the same as with steam engines. The space occupied by the new type is only about 60 percent of that required by steam engines and boilers, and there would be large savings in weight. This can only be regarded as a sketch design, as it goes very far in power beyond any actual installation, but it is suggestive. For British submarines Messrs. Vickers have made engines up to 300 brake horse power, and the Daimler Company has made engines of equal power. In the small swift launches so far built, it is claimed that the propelling apparatus and fuel for six hours weigh only from one-sixth to one-tenth as much as for a good condensing steam engine and boiler, suitable for boats and capable of developing the same power. There is a great saving in space occupied by propelling apparatus, a large increase of accommodation, and much enlarged power of covering distance. In a good steam engine fitted in boats from 2 to 2 1-2 pounds of coal per horsepower hour are consumed; with oil about one-third that weight suffices.

Besides oil engines and spirit engines, gas engines (using gas made from solid fuel by producers) are under consideration for marine purposes. The boiler and its contained steam and water would disappear in favor of producers, cheaper qualities of coal would suffice for making gas, and it is anticipated that large economies in cost, as well as weight and space, would be effected. Practical steps are being taken to give effect to this idea.

Claims are also made that gas turbines will be brought into use, and the writer has had statements made to him recently that engines of this class of considerable size have been made and worked. As yet, however, there seems no sufficient evidence that the great difficulties inherent in the system have been overcome.

OIL FUEL.

British experience with oil fuel as a substitute for coal has not equalled that of oil-producing countries. Experiments have been made, however, on a considerable scale for marine purposes; the Admiralty having recently resumed the investigation which was begun many years ago and then suspended, while many merchant ships have been fitted for oil fuel. Satisfactory arrangements for efficiently burning oil fuel are now available for British ships, and every one recognizes the enormous saving on shipping oil and transporting it to the furnaces, as well as the advantages obtainable from its greater calorific value. The practical difficulty in Great Britain, however, and probably the crucial point in all countries, is that of adequate supplies of oil at reasonable prices. Statements have repeatedly been made that this difficulty has been overcome, but it remains as real now as it was ten years ago. Coal is holding its own for ship propulsion except in a few special trades.

BRITISH WARSHIPS IN 1904.

The latest type of battleship for the Royal navy, of which the first examples are now approaching completion, is the *King Edward* class, designed by the writer in 1901. These vessels are 425 feet long, 78 feet broad and 16,350 tons displacement. Their engines develop 18,000 horsepower, and the guaranteed speed was 18 1-2 knots for 8 hours. The first vessel tried (the *Commonwealth*) was ballasted to about 2 inches deeper than the designed draft of 26 feet 9 inches, and with 18,500 I. H. P. she exceeded 19 knots. On a 30-hour trial, with 12,770 I. H. P., she attained 17.9 knots, and the coal consumption was 1.68 pounds per horsepower hour. The disposition of armor differs from that of the *Majestic* of 1894 in many respects. The after end of the armored citadel wraps round the base of the after barbette, as in the *Majestic*, and the top of this armor is at the level of the main deck; but this depth of armored side (about 14 1-2 feet) is continued to the bow, maintaining its full thickness until the fore barbette has been passed, and then being gradually diminished to 3 inches at the stem. Abaft the citadel the water-line region is protected by thin side armor. There are two protective decks: one, nearly horizontal, about 9 feet above water, and the other, curved transversely (as in *Majestic*), reaching to the lower edge of the side armor. The four 12-inch guns are mounted in pairs in barbettes and carried on revolving turn-tables with armored shields. Between the barbettes, on the main-deck, is a central battery with 7-inch armor on the sides, containing ten 6-inch quick-firing guns. On the upper-deck, four 9.2-inch guns are mounted in four separate turrets. The side armor is 8 and 9 inches, the barbette armor 12 inches; the total weight of armor and backing is 4,800 tons. All the armor is of Krupp quality. The new feature in the armament is the use of 9.2-inch guns to supplement the 6-inch guns; a change rendered necessary by the fact that improvements in armor manufacture had made the protection given to secondary armaments in recent battleships superior to the penetrating power of 6-inch guns at the long ranges now adopted. Since the writer's retirement three other vessels have been ordered from this design, so that there are now eight of the class building. The hull costs about £400,000, and the armor about the same amount. Propelling and other machinery cost nearly £220,000; gun mountings and torpedo tubes £215,000; and the total cost per ship exceeds £1,337,000, exclusive of armament and ammunition. Fully equipped each ship represents over 1 1-2 millions sterling.

The interval of ten years has raised the displacement of a first-class battleship from 15,000 to 16,350 tons and the cost from £820,000 to £1,337,000, excluding armament. The speed has been increased more than a knot, and great improvements have been made in armor, guns, and gun mountings; the defensive and offensive power being considerably increased.

These features of change in design extend to all classes of modern warships, but it is not possible to illustrate the tendency by references to many classes. Taking armored cruisers, the *Drake* may be contrasted with the *Powerful*, a protected cruiser of 1894. They are practically of the same length, breadth, and displacement; and, as both classes were designed by the writer, it is obvious that personal skill had nothing to do with the result. Owing to changed conditions, it was desired, in the *Drake*, to develop armor protection, to increase speed and to make the armament more powerful as compared with her predecessor. This involved a redistribution of the weights making up the displacement. By changes in freeboard and in superstructures, the abolition of the heavy and costly wood and copper sheathing, which had been thought necessary in the *Powerful* for her special service, and the acceptance of less thicknesses of deck protection, a considerable saving in weight was effected. With the weight thus saved it was possible to give the *Drake* 6-inch side armor and 5-inch casemates and shields; to increase the armament by two 6-inch guns; and to add about 36 percent to the engine power, with a gain of fully a knot in estimated speed. On the other hand, the maximum coal capacity was reduced from 3,000 to 2,500 tons. The *Drake* class cost rather more than one million sterling, exclusive of armament, the armor representing a considerable proportion of the total, and the more powerful engines involving large outlay. The *Powerful* cost less than £700,000; the only thick armor carried was that on casemates and shields, and the protective deck was comparatively inexpensive.

Of the numerous classes of cruisers, both armored and protected, which now form part of the Royal navy, it is impossible to speak on this occasion. New developments are continuously taking place, amongst the most notable in recent years being the construction of a great flotilla of destroyers and a few scouts, to which reference has been made previously. Submarine vessels of the Holland type have been added to the fleet, and auxiliaries to

*Paper read before the International Engineering Congress at St. Louis, October 8, 1904.

fleets—repair and depôt ships, water tanks and colliers—have been built, or are contemplated. According to the latest Parliamentary returns, on March 31, 1904, the British navy included the following vessels, built and building, or to be commenced in 1904: battleships, 67; coast-defence ironclad, 1; armored cruisers, 45; protected cruisers and scouts, 116; unprotected cruisers, 8; torpedo vessels, 32; destroyers, 160; torpedo boats, 91; submarines, 29; a grand total of 549 vessels.

The examples above given for certain classes explain the enormous increase of expenditure on war fleets which has occurred in recent years. The reconstruction of the Royal navy, which took place during the writer's period of office, under arrangements made in successive programmes, involved an expenditure, from 1885 to 1902, of 88 1-2 millions sterling, and the 245 ships of all classes, for which the writer was the responsible designer, had an aggregate value of about 80 millions sterling, exclusive of armaments and stores, or about 100 millions fully equipped. The capital value of the fleet in the same period was about trebled, and became 100 millions, exclusive of armaments. At the end of the French war (1813) the corresponding capital value was 10 millions; in 1860 it was under 18 millions; in 1878, 28 millions. The maintenance of supremacy at sea is a very costly undertaking in modern times, and the responsibility of the naval architect grows continuously.

STABILITY.

In regard to the developments in the theory of naval architecture during the last ten years little need be said. The most marked change has been in the direction of wide extensions of scientific procedure in the designing of ships.

The scientific principles underlying calculations for the stability of ships, and the methods of calculation, have been long determined. For more than a century the use made of these methods was very limited. Their practical application on a large scale received an enormous impetus from the inquiry which followed the loss of H. M. S. *Captain* in 1870. The chief advances made since that date have been the device and perfection of integrating machines, by which great economies of time and labor have been effected, and the accumulation of masses of experimental data for ships of various classes. Formerly, calculations for the stability of merchant ships were the exception. The introduction of novel types and the more general adoption of scientific procedure in design has, in recent years, led to widespread experimental investigation, and to detailed calculations for all classes of merchant ships. Indeed the practice for merchant ships is now very similar, in this respect, to that previously followed for war ships.

One feature of great importance may be mentioned, which has been developed, during the last ten years, by the introduction, in battleships and armored cruisers, of heavy guns with strong armor protection, placed high above water. Personally, the writer recognized the fact, from the first, that such a vertical distribution of weights must produce an unusually high position of the center of gravity, in proportion to the total depth of the ships; and that, even with high freeboard, there would be only moderate ranges in the curves of stability, as compared with preceding battleships and cruisers of high freeboard. For preceding high freeboard types, either for war or commerce, it had been usual and proper to assume that, if the metacentric height was sufficient, there would be ample range; but under the new conditions it seemed clear to the writer that this could no longer be true.

Calculations made (about 1893-94) for battleships of the *Majestic* class and for the cruisers *Powerful* and *Terrible* confirmed this anticipation, and showed that, notwithstanding the very high freeboard, the range of stability was only about 60 degrees. At present one of the most critical questions to be faced by warship designers is that of securing adequate range of stability with heavy loads of armor and armament, which are required to be carried at great heights. In fact, so far as the character and range of curves of stability are concerned, the problems now arising are similar to those which were dealt with, for vessels of moderate freeboard, by scientific members of the committee on designs for ships of war appointed by the British Admiralty in 1871. Their investigations for vessels of the breastwork-monitor type have consequently acquired fresh interest. Although these investigations were conducted on scientific lines, and secured the collaboration of men like Froude, Rankine, and Lord Kelvin, the solutions reached and the practical rules laid down were avowedly, and from the nature of the case could be, only approximate. To treat them as if they were exact and complete solutions, would be distinctly contrary to the views of the investigators themselves.

When guns of heavier calibers and greater lengths have to be mounted at considerable heights above water, in order to increase fighting efficiency, there is a temptation to accept limitations in range of stability, which may be of serious import in war ships exposed to damage in action. Increase in breadth, of course, enables adequate metacentric heights and some addition to range to be obtained, even under the conditions described. But everyone familiar with the subject is aware that while increase in beam, and consequent increase in metacentric height, produces some addition to range of stability, the gain is relatively small if the center of gravity is high in the ship. On the other hand the period of oscillation is rapidly diminished by increase in metacentric height, and quicker rolling is accompanied by loss of steadiness at sea, under critical conditions of wave motion. It is not overlooked that in modern types the greater area of armor protection and the greater vertical depth of armored sides, make ships much less liable to rapid diminution of their stability when exposed to gun fire in action; so that there is reason for accepting a less range of stability in the intact condition than with preceding ships of inferior protection. But the writer's conviction is that the question of range of stability, under existing conditions, does not always receive the attention it deserves; and that the tendency which now prevails to increase the caliber of guns in secondary armaments and to place them on upper decks, makes it most important to thoroughly investigate the conditions of stability for each design before accepting the desired armaments and protection.

As to merchant ships the cases requiring closest study are those incidental to cargo steamers carrying great loads and very diverse cargoes. The designer has no control over the character or stowage of the cargoes. All that he can do is to assume certain conditions as being the worst likely to occur in practice, and to provide for them. Great attention is now given to such problems, and scientific procedure in regard to them is general. Legislation in regard to maximum load lines in British ships has greatly assisted designers in their work, by fixing one important feature. In essentials, that legislation remains practically in the same form as it stood ten years ago.

ESTIMATES OF SPEED AND ENGINE POWER.

In this section of ship design it cannot be said that the last ten years have added any important novelty in procedure; but there has been a large extension in the application of accepted methods based upon scientific investigation. The necessity for progressive steam trials and their thorough analysis is now generally recognized. Model experiments in specially constructed tanks are much more common, and are admitted to be essential to the complete study of designs for ships of unusual form or unprecedented speed. Leading shipbuilding firms are establishing their own experimental tanks; and a movement is now in progress for adding to these private establishments in England, a tank specially devoted to general research work on problems incidental to the movement of solid bodies through water and the efficiency of propellers. Experience to date establishes beyond question the opinion, long held by those most familiar with the subject, that, apart from experimental inquiry, mathematical investigation will never be capable of dealing conclusively with the causes influencing the efficiency of screw propellers, or the selection of the propellers best adapted to a particular ship. Fresh weight has been given to this argument by the introduction of rotary engines

with high speed of revolution, demanding propellers of unusual dimensions in relation to the power applied. All these matters are now receiving attention. They cannot but have a marked effect upon the economy and efficiency of future steamships both for war and for commerce. It is unnecessary to reproduce many facts illustrating these general conclusions; a single example must suffice. The *Drake* class of armored cruisers was designed for 23 knots speed on an eight hours' trial. The writer's estimate was that, if the choice of propellers was successful, the actual speed would be about 23 1-2 knots. On the trial of the *Drake*, with the screws first fitted, 30,600 I. H. P. was developed and the corresponding speed was 23.05 knots. These screws had been selected on the basis of experiments with model screws made in the Admiralty tank. From progressive trials, however, it was ascertained that at the higher speeds the slip rapidly increased, and it was obvious that greater blade area was required. New blades were made of the same diameter, rather less pitch, and about 38 percent greater area. With 31,400 I. H. P. the speed attained with these propellers was 24.11 knots, corresponding to 24 knots for 30,600 I. H. P., and 23.05 knots for 26,000 I. H. P. That is to say, the change of propeller blades produced an economy in power of 15 percent at 23 knots, and enabled a knot greater speed to be realized at full power. Equally striking examples might be given from experience with other classes, and particularly with destroyers, all of them indicating the necessity for further experiments on a large scale to supplement model experiments with screws, and to furnish a proper scale of comparison.

One of the latest and most striking examples of the value of tank experiments has been furnished in connection with the design of the great Cunard steamships now building, which are to be capable of maintaining an average speed of 24 1-2 knots across the Atlantic in fair weather. Here, in association with the selection of the most suitable dimensions and form, and the settlement of novel problems arising from the use of four shafts and propellers driven by steam turbines, it would have been impossible to proceed with confidence apart from tank experiments on models. By the courtesy of the Admiralty these experiments have been exhaustively completed with the greatest advantage to the owners and designers of the ships.

In conclusion it is necessary to repeat the statement that within the limits of this paper anything like a comprehensive survey of the progress of British shipbuilding since 1894 is impossible. As one who has been in close touch with fellow workers in all departments, and who has himself been actively engaged, it has been the writer's endeavor to note the chief features in a remarkable period, and to indicate present tendencies. He trusts that the review, imperfect as it is acknowledged to be, will have some interest for the members of this congress.

Havana Auto Races.—It is announced that President Palma has issued a permit for automobile and auto-boat races, to take place in Havana during the month of February. The course for automobiles will be 204 miles on the round trip, being on the San Cristobal road, and will occur on February 8, 9, 10, and 11. The auto-boat tests will take place on the 4th of February and will probably consist of three races. On the last day of the automobile race a tour will be made of the principal points of interest in the province of Havana, the affair ending at night with the parade and carnival.

OBITUARY.

We have to record the death, on the 28th of December, of Thomas F. Secor, at the age of 89. In his day Mr. Secor was one of the best known shipbuilders in the country. He was the builder of the Mare Island navy yard, at San Francisco, California, and the Pensacola navy yard at Pensacola, Florida. During the Civil war he was responsible for the construction of some of the monitors used by the United States, as well as other ironclads and dry docks.

POWERING SHIPS.

BY EDWIN CERIO, NAVAL ARCHITECT.

PART II.

IV. THE RESISTANCE OF SHIPS.

It is an inexplicable but undoubted fact, that quite a number of theories on the resistance of ships have up to 1870 enjoyed much credit, though founded on quite erroneous interpretations of the phenomena of a vessel moving in water. Scott Russell's theory, to mention one, was founded on the assumption that a ship when in motion performed an *excavation* in the water, and that therefore to work her way she had to lift from the center of buoyancy to the surface a mass of water equal to the displacement. If this were true the propulsion of vessels would need an enormous amount of force, but, as Froude observed in his address to the Royal Institute, May, 1876: "It is a point worth noticing what an exceedingly small force, after all, is the resistance of a ship compared with the magnitude of the phenomena involved." Froude further quotes the example of the frigate *Shah*, and says that scarcely anyone seeing her steam at full speed "would be inclined at first sight to credit, what is nevertheless a fact, that the whole propulsive force necessary to produce that apparently tremendous effect is only 27 tons, in fact, less than one two-hundredth part of the weight of the vessel."

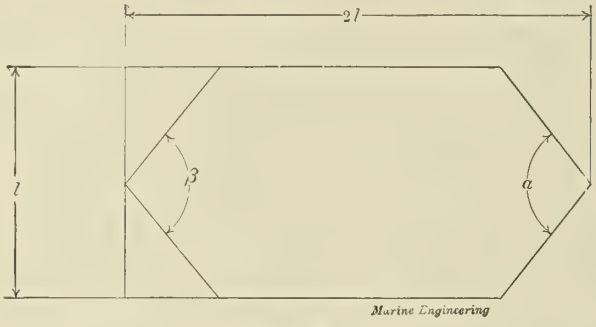


FIG. I.

Up to the year 1870 nearly all of the numerous formulæ given for calculating the resistance of ships were derived from the expression:

$$W = KAV^2,$$

where A is the midship section area, V the speed, and K a coefficient. Though this formulæ is no more employed in our day to determine the resistance of ships, it may be of some use in the case of prismatic vessels such as pontoons, barges, etc., as it is founded on a sound basis; in fact, if A is the area of a plane moving in a fluid with a certain speed V, and if δ is the density of the fluid, in the unit of time, the plane A will have removed a volume of fluid AV, which will have performed an amount of mechanical work proportional to:

$$\tfrac{1}{2}\,\delta\,AV \times V^2,$$

if this work be equalled to that of the resistance W, $W\,V \sim \tfrac{1}{2}\,\delta\,AV \times V^2$ and

$$W \sim \tfrac{1}{2}\,\delta\,AV^2$$

To take into consideration all the other causes which influence the resistance when the body moving is no longer a plane, the coefficient K has been introduced, thus obtaining the formulæ $W = KAV^2$. Many authors have given a series of values of K which, unhappily, differ much from one another. Accurate researches by Froude have led to the value:

$$K = 11.3.$$

When the formula $W = KAV^2$ is to be used in practice the following values of K may be found of great help for vessels of prismatic form:

When the ratio of length to breadth is from 4 to 6, $K = 10.3$.
In the same conditions but with extremities sharpened $K = 7.2$.
Prismatic vessels ship-shaped at ends, $K = 1$.

The following remarks may also prove useful when considering such vessels:

A prismatic vessel having a square water-line section moving in the direction of the diagonal meets a resistance 1.221 times that which it meets when moving in the direction of the sides.*

The following data show the influence of angles afore and abaft of a prismatic body on its resistance in the water.**

If, in Fig. 1, when $\alpha = 180$ degrees and $\beta = 180$ degrees the resistance is $W = 1$, for

$\alpha = 150$ degrees	$W = 0.96$	$\beta = 150$ degrees	$W = 0.95$
$\alpha = 90$ "	$W = 0.56$	$\beta = 90$ "	$W = 0.90$
$\alpha = 60$ "	$W = 0.45$	$\beta = 60$ "	$W = 0.86$
$\alpha = 30$ "	$W = 0.41$	$\beta = 30$ "	$W = 0.84$
$\alpha = 15$ "	$W = 0.40$	$\beta = 15$ "	$W = 0.82$

THE LAW OF KINEMATIC SIMILITUDE.

To study the resistance of ships by means of their models, it is necessary that the dimensions of ship and model be in a certain ratio α, and that if V is the speed of the vessel and v that of its model,

$$V = v\sqrt{\alpha}.$$

When the speeds V and v satisfy this condition they are called "corresponding speeds." Corresponding speeds produce similar wave configurations. "When two similar vessels move with corresponding speeds, the resistances met by these vessels are proportional to the cubes of their linear dimensions; that is to say, their ratio is equal to the ratio of the vessel's volumes"; on this law, which is Newton's law of kinematic similitude, is founded the study of ships' resistance by means of their models. Were this law true for the whole of the resistance which a vessel meets in water, it would be enough to multiply the model's resistance, and its speed respectively by the third power (α^3) and the square root ($\sqrt{\alpha}$) of the ratio of similitude to obtain the resistance of the ship at the corresponding speed. But the law of similitude does not apply to all the causes which affect the resistance of the ship, and it proves true only for that part of resistance which is due to wave configuration; the other part of the ship's resistance, that is, the skin resistance, must be investigated separately.

The method originally introduced by Froude to apply the law of similitude to ship resistance is the following: By means of tank experiments the total resistance w of the ship's model is determined, and from this the skin resistance w_s is subtracted, thus obtaining the residual resistance w_r. This residual resistance of the model multiplied by α^3 gives the residual resistance W_R of the ship; the ship's skin resistance W_S is calculated separately. The residual resistance comprehends also the secondary resistances, and it may be assumed that these, like the resistance due to wave configurations, also follow the law of similitude. The total resistance may be expressed by:

$$W = W_R + W_S,$$

where:

$$W_R = (w - w_S)\alpha^3;$$

therefore we may also write:

$$W = (w - w_S)\alpha^3 + \frac{W_S}{\alpha^3}\alpha^3,$$

or:

$$W = \left(w - w_S + \frac{W_S}{\alpha^3}\right)\alpha^3 = W_1\alpha^3,$$

having made

$$w - w_S + \frac{W_S}{\alpha^3} = W_1.$$

The law of variation of resistance in proportion with speed may be represented by a curve in which the speeds are taken as abscissæ and the resistances as ordinates. Let the curve (w) Fig. 2, be the diagram of the total resistance of the model; we can also construct a curve (W) with the same abscissæ and having for ordinates the values of:

$$w - w_S + \frac{W_S}{\alpha^3}$$

for the different speeds, as w, w_s and $\frac{W_S}{\alpha^3}$ are known quantities. If the ordinates of the curve W are divided by α^3 and the abscissæ by $\sqrt{\alpha}$, that is, if we read the resistances on a scale $(w \div \alpha^3)$ and the speeds on a scale $(V \div \sqrt{\alpha})$, the curve W will represent the law of variation of the total resistance of the ship.

This original way of applying Froude's theory is very simple, but it does not allow us to compare to the same model a series of ships whose linear dimensions bear to those of the model different ratios of similitude α_1 α_2 α_3 . . . because when the dimension length varies, the values of $\frac{W_S}{\alpha^3}$ also vary, and the same curve of one model could not serve to obtain the resistance of all similar ships. Another method, to be described at a later point, is therefore employed when the resistance of ships is to be determined by means of models.

SKIN RESISTANCE.

The principal feature of modern researches on the resistance of ships is the introduction of a cause of resistance which had been, in the old methods, completely left aside; this is the friction generated by contact of the outside plating with the water, and which causes the skin resistance.

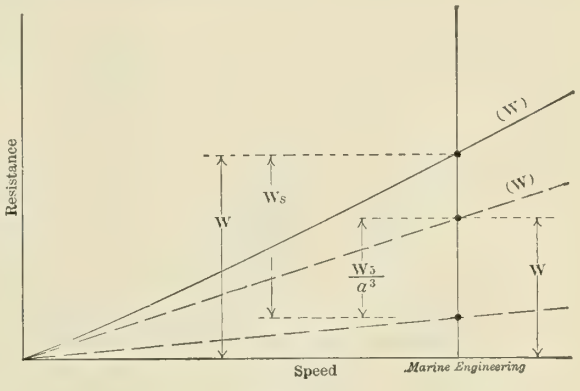


FIG. 2.

The water in immediate contact with the ship assumes irregular movements, and a belt of water in such conditions will surround and accompany her in her progress, continually drawing a certain amount of energy to maintain the perturbed state of the liquid molecules. This amount of energy, which continually passes from the ship to the water, is transformed into heat and performs therefore no useful mechanical work; it requires, however, a certain amount of power, which is the power due to skin resistance.*

The most important geometrical element necessary to determine skin resistance is the *wetted surface* of the ship; strictly speaking, the *reduced wetted surface* ought to be employed, but the actual surface is always considered, and is, practically, the same in most cases. The following formulæ give fairly accurate values of the wetted surface:

$$S = 15.6\sqrt{D.L}$$
$$S = L\,(1.7H + \delta B),$$

Where:

S is the wetted surface in square feet,

*Gareis—Wasserwiderstand des Schiffes.

**Scribanti—Lezioni di teoria della nave.

*I have found it appropriate, in the following remarks, to repeat only that part of the theory of skin-resistance which is indispensable for practical use, when powering ships. This theory is dealt with, by Prof. W. F. Durand, with great clearness and competence, and all those especially interested in the subject may profitably consult his book, "Resistance and Propulsion of Ships."

D the displacement in tons,
L the length in feet,
H the draft in feet,
B the beam in feet,
δ the block coefficient.

These formulæ may be always applied in practice; when particular accuracy is required, other means of determining the wetted surface with greater exactness are always at hand.

Calling:
W_s the skin resistance in pounds,
f the coefficient of skin resistance,
S the area of wetted surface in square feet,
n an index for the variation of resistance with speed,
the general formula by means of which skin resistance may be determined is:

$$W_s = f\,Sv^n.$$

The coefficient f depends mostly on the nature of the wetted surface, on the density of the water, and varies indirectly with the length of the ship. It is affected by temperature, but only slightly, and depth and pressure of water have no influence upon it.

The exponent n is influenced by the roughness of the outside plating, and by the length of the ship.

Table V contains the values of f and of n, to be adopted in each case.

Experiments carried out at Spezia, Italy, to ascertain the influence of paint on resistance seem to prove that the quality of paint, practically, does not affect skin resistance. If f is the coefficient of skin resistance when the outside plating is freshly painted and perfectly smooth, the coefficient when (1) the outside plating is not painted is 1.4 f; (2) the ship has a foul bottom is 2 f; (3) the ship is covered with incrustations is 5 f.

TABLE V.—VALUES OF THE COEFFICIENT OF SKIN-RESISTANCE AND OF THE INDEX FOR THE VARIATION OF RESISTANCE WITH SPEED.

For Ships* (in Salt Water).					For Paraffin Models (in Fresh Water).		
Length of Ship, Feet.	According to Froude. Values of:		According to Tidman. Values of:		Length of Model, Feet.	According to Tidman. Values of:	
	f.	n.	f.	n.		f.	n.
8	0.01197	1 825	.....	.....	2	0.01176	1.94
9	0.01177	1,825	.....	.....	3	0.01123	1.94
10	0.01161	1,825	0.01124	1.8530	4	0.01083	1.94
12	0.01131	1,825	.....	.....	5	0.01050	1.94
14	0.01106	1,825	.....	.....	6	0.01022	1.94
16	0.01086	1,825	.....	.....	7	0.00997	1.94
18	0.01069	1,825	.....	.....	8	0.00973	1.94
20	0.01055	1,825	0.01075	1,8490	9	0.00953	1.94
25	0.01029	1,825	.....	.....	10	0.00937	1.94
30	0.01010	1,825	0.01018	1.8440	10.5	0.00928	1.94
35	0.00993	1,825	.....	.....	11.0	0.00920	1.94
40	0.00981	1,825	0.00998	1,8397	11.5	0.00914	1.94
45	0.00971	1,825	.....	.....	12,0	0.00908	1.94
50	0.00963	1,825	0.00991	1.8357	12.5	0.00901	1.94
60	0.00950	1,825	.....	.....	13.0	0.00895	1.94
70	0.00940	1,825	.....	.....	13.5	0.00889	1.94
80	0.00933	1,825	.....	.....	14.0	0.00883	1.94
90	0.00928	1,825	.....	.....	14.5	0.00878	1.94
100	0.00923	1,825	0.00970	1.8290	15.0	0.00873	1.94
120	0.00916	1,825	.....	.....	16.0	0.00864	1.94
140	0.00911	1,825	.....	.....	17.0	0.00855	1.94
160	0.00907	1,825	.....	.....	18.0	0.00847	1.94
180	0.00904	1,825	.....	.....	19	0.00840	1.94
200	0.00902	1,825	0.00944	1.8290	20	0.00834	1.94
250	0.00897	1,825	0.00933	1.8290	....	.....	....
300	0.00892	1,825	0.00923	1.8290	....	.....	....
350	0.00889	1,825	0.00916	1.8290	....	.....	....
400	0.00886	1,825	0 00910	1.8290	....	.....	....
450	0.00883	1.825	0.00906	1.8290	....	.....	....
500	0.00880	1,825	0.00904	1.8290	...	.....	....
550	0.00877	1,825	...	.....	....	.....	....
600	0.00874	1,825	.....	.....	....	.....	....

*The values of f and n are for steel ships with the bottom perfectly clean and well painted.

N. B.—For intermediate values of the length, the table is used with interpolation.

WAVE RESISTANCE AND ECONOMICAL SPEED.

The motion of a ship causes the surrounding water to assume what is called a *wave configuration*. To maintain this wave configuration it is necessary that a certain amount of energy be continually transmitted from the vessel to the liquid particles which surround it, or in other words, that a part of the propelling power be employed to neutralize the work of wave resistance. The mass of water which forms a wave does not advance, but the wave configuration does, and in advancing it abandons half of the energy which it required for its *formation;* therefore, when the vessel moves, producing a wave configuration, it is necessary that the part of the energy required to maintain the configuration only, and not that part required for the formation of the wave, should continually be furnished, in the shape of horse power due to wave resistance.

The following remarks apply in a general way to wave resistance:

Wave resistance depends upon speed, and, at a given speed, upon the length of the ship. For a given length there must be a certain speed, above which the resistance begins rapidly to increase, and this speed is proportional to the square root of the ship's length. This fact may be observed in Fig. 3, where some typical curves of resistance are shown for four vessels; the coefficient δ, given for each, is the block coefficient. It is of great interest to more closely investigate the characteristics of the above-mentioned speeds, which, though very seldom considered in practice, have an importance which ought not to be overlooked by any ship designer.

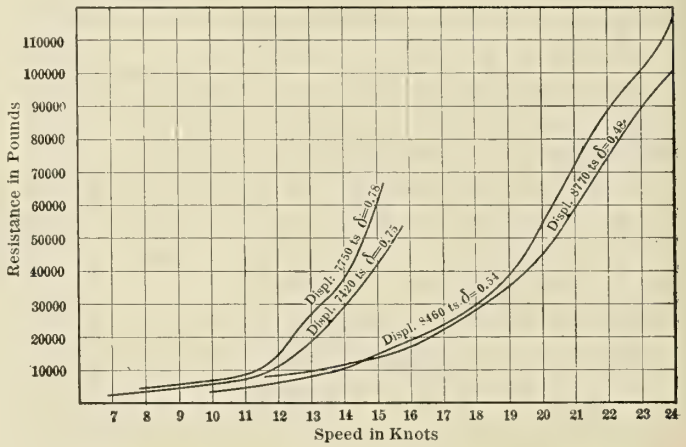


FIG. 3.

Many shipbuilders have a vague idea of what is generally called the *economical speed of a ship,* and its relation to the ship's length, associating this idea with the notion of the *smallest amount of power* required to propel a vessel in given conditions. Mr. E. Tennyson-D'Eyncourt, in a very important paper read before the Institution of Naval Architects,* analyzing the trial results of many ships, gives a method of defining the *limiting speed;* he says every ship has an appropriate limit of speed, beyond which it is uneconomical to attempt to drive her; this would lead to define economical speed as the limit beyond which every increase in speed requires a disproportionate increase in power. But this and other abstract definitions convey only a conventional idea of the influence which form and dimensions have on wave resistance. A closer study of this most important phenomenon, viz., the existence of a certain characteristic speed for every ship, will give a more concrete notion of a fact which ought always to be kept in view when powering ships.

To avoid the elaborate discussion which leads to a very simple expression of wave resistance, I will recall this expression and omit the demonstration, as it is only from the final result that I intend to draw the conclusions which will give a clear idea of what has to be understood under "economical speed." Calling:

W the wave resistance,
b the mean value between the breadth of a wave forward and that of a wave aft,

*"On the Limits of Economical Speed."

h_1 and h_2 respectively the heights of these waves,

$h'_1 = Kh_1$ the height of the forward wave when its configuration has reached the point at which the after wave is formed, K being a constant,

m the ratio between the distance apart of the two wave crests, and the length of the ship,

L the length of the ship,

V the speed,

g the acceleration due to gravity,

α and β two constants resulting from the equations,

$$h_1 = \alpha V^2 \qquad h_2 = \beta V^2,$$

the expression referred to of wave resistance may be put under the form:

$$W = b\left\{\alpha^2 + \beta^2 + 2K\alpha\beta \cos\frac{gmL}{V^2}\right\} V^4.$$

We shall consider only the factor $\cos \frac{gmL}{V^2}$, in which g and m are constants, and therefore L and V^2 are the only terms which affect the wave resistance; the following discussion, though aiming to give a practical and clear idea of what is generally called economical speed, is purely theoretical, and its conclusions cannot be applied, so to speak, numerically. The factor $\cos \frac{gmL}{V^2}$ may be written $\cos \frac{gm}{G^2}$ where the constant $G = \frac{V}{\sqrt{L}}$ and a diagram

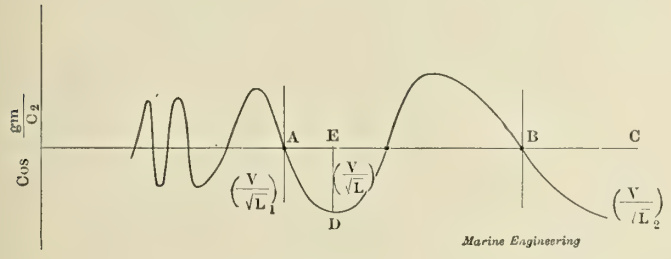


FIG. 4.

constructed with the values of $\cos \frac{gm}{G^2}$ as ordinates and those of G as abscissæ will have the form shown in Fig. 4. There are some values of V and L which will make the factor $\cos \frac{gmL}{V^2}$ negative and others which make it positive.

The expression of wave resistance shows clearly that the resistance will be smaller, within certain limits of speed, when $\cos \frac{gmL}{V^2}$ has the greatest negative value; in the region graphically limited by A and B, in Fig. 4, between the values $\left(\frac{V}{\sqrt{L}}\right)_1$ and $\left(\frac{V}{\sqrt{L}}\right)$ of the abscissæ there is a value $\left(\frac{V}{\sqrt{L}}\right)$ for which $\cos \frac{gm}{G_2}$ has a maximum negative value; *in this region,* the length of the ship being L, it may be said that V is the economical speed, because the wave resistance is here the smallest for values of G between A and B. In the formula of resistance the value of the constant K is proportional to the speed. The value of the latter may become so great that to avoid a disproportionate increase of resistance it may be advisable to increase L; these *theoretical* conclusions are confirmed by practice. Mr. Tennyson-D'Eyncourt deduces many important conclusions in his paper on the limits of economical speed, the following being those of more practical value when estimating horsepower. A relation between the coefficient of fineness, the length of parallel body, and the limits of economical speed is given in the following table:

TABLE VI.

Coefficient of Fineness.	Parallel Body as Percentage of Total Length of Ship at the W. L.	Limiting Economical Speed in Knots as Percentage of $\sqrt{L}$ in Feet.
0.5	0	100
0.6	14	86
0.7	30	68
0.8	49	48

The I. H. P. at the limiting speed is varying as the 4th power of the speed, and varies in increasing ratio until at about 12 percent above the limiting speed it is varying as the 7th power of the speed.

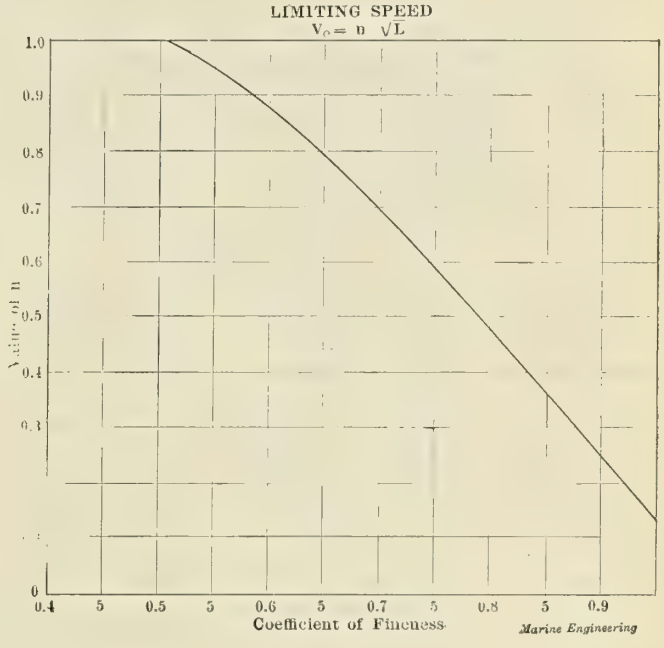


FIG. 5.

The wave horsepower varies as V^7 at the limiting speed and as V^{10}, or sometimes at a higher power of V, at 12 percent above the limiting speed; at this point, viz., at 12 percent above the limiting speed, the wave horsepower is approximately equal to the skin horsepower. The values of the Admiralty's coefficient, $C = \frac{D^{\frac{2}{3}} V^3}{IHP}$ are as follows:

TABLE VII.

Length of Ship in Feet.	Values of C at the Limiting Speed.	Values of C when Skin HP = Wave HP.
200	225	188
300	256	207
400	278	224
500	295	237
600	310	246

In analyzing the trial results of ships, Mr. Tennyson-D'Eyncourt has assumed the performance:

$$E = \frac{EHP}{IHP} = 0.5.$$

In a study of this nature it is thought advisable not to rely upon arbitrary coefficients of performance, and to consider, whenever it is possible, the effective horsepower, applying afterward, in every single case, the particular value of the coefficient E, which appears to be best suited. I have analyzed the results of experiments carried out with models of actual ships. Table VIII contains, besides the geometrical characteristics of a number of ships of different types, the economical speed and the E. H. P. due to skin resistance and wave formation at different speeds given in

TABLE VIII.—RELATION BETWEEN E. H. P. AND THE LIMITING ECONOMICAL SPEED.

No.	Description of Vessel.	Length, Ft.	Breadth, Ft.	Draft, Ft.	Wetted Surface, Square Ft.	Coefficient of Fineness, δ.	Displacement, Tons.	Effective Horse Power at Speeds Given in Percentage of the Economical Speed. 60 EHP$_W$	60 EHP$_F$	70 EHP$_W$	70 EHP$_F$	80 EHP$_W$	80 EHP$_F$	90 EHP$_W$	90 EHP$_F$	100 (Economical Speed) EHP$_W$	100 (Economical Speed) EHP$_F$	110 EHP$_W$	110 EHP$_F$	120 EHP$_W$	120 EHP$_F$	130 EHP$_W$	130 EHP$_F$	Economical Speed, Knots.
1	Cargo-boat No. 1.	360.9	43.4	22.6	25,890	0.780	7,750	...	...	94	175	120	254	155	360	210	480	300	.....	.....	.....	.....	.....	10.
2	Cargo-boat No. 2.	360.9	43.4	22.6	25,490	0.750	7,420	50	156	83	242	131	352	185	485	272	645	520	835	940	1,070	1,360	1,750	11.2
3	Passenger steamer No. 1.	442.9	53.6	22.9	29,500	0.540	8,460	238	930	487	1,400	950	2,080	1,750	2,900	3,480	3,900	6,200	5,100	8,800	6,500	11,500	8,150	20.
4	Passenger steamer No. 2.	442.9	53.56	22.95	30,500	0.563	8,880	160	890	430	1,400	880	2,040	1,740	2,850	3,850	3,850	9,000	5,150	.....	.....	.....	.....	19.6
5	Passenger steamer No. 3.	492.1	55.8	22.9	34,090	0.485	8,770	450	1,480	840	2,300	1,640	3,350	3,340	4,700	6,400	6,350	9,300	7,850	.....	.....	.....	.....	22.7
6	Passenger steamer No. 4.	442.9	55 8	22.9	29,800	0.600	9,670	186	740	395	1,140	650	1,750	1,270	2,410	2,260	3,250	3,620	4,120	6,550	5,500	9,200	7,000	18.6
7	Passenger steamer No. 5.	288	36.58	13.78	11,680	0 490	2,150	56	236	113	365	239	535	475	760	810	1,000	1,370	1,340	2,300	1,660	4,700	2,200	17.20
8	Passenger steamer No. 6.	288	37.2	13.78	12,050	0.530	2,209	20	225	68	330	183	480	385	670	920	920	1,490	1,200	2,160	1,500	.....	.....	16.50
9	Military transport.....	209.9	37.7	11.66	8,700	0.525	1,380	74	106	111	164	173	236	266	321	455	435	760	580	980	680	1,650	950	14.2
10	Battleship *Re Umberto*....	400.3	76.3	24.44	34,100	0.520	12,650	425	1,030	815	1,580	1,460	2,320	2,600	3,240	5,650	4,400	9,000	5,800	12,500	7,400	.....	.....	19.60
11	Cruising ship....	395.1	50.9	19.68	16,800	0.494	4,300	180	360	306	575	525	845	935	1,165	1,660	1,590	3,100	2,120	.....	.....	.....	.....	17.6
12	Cruiser *Piemonte*.....	304.1	38.2	15.28	13,220	0.495	2,518	...	286	...	440	....	640	306	915	515	1,410	1,000	1,600	2,240	2,040	5,100	2,600	17.6

percentages of the economical speed V ; this has been each time determined by means of the formula:

$$V_E = n\sqrt{L}.$$

where L is the length at the water line in feet, and n a coefficient obtainable from the diagram shown in Fig. 5.

With this rather long digression on economical speed, it is intended to call attention to a problem which is intimately connected with the general problem of powering ships—the problem of finding the most favorable relation between the forms of vessels and the speeds they have to attain under fairly economical conditions.

Practically the elements which mostly influence wave resistance are displacement, length, and speed; calling p an experimental coefficient, according to Taylor the wave resistance may be expressed by the formula:

$$W_W = p\,\frac{D^{\frac{2}{3}}}{L}\,V^4.$$

W is the resistance in pounds,
L the length of the ship in feet,
D the displacement in tons,
V the speed in knots, and
p a coefficient which may be assumed:

0.365 for vessels with very fine lines and a great ratio of length to breadth;

0.400 for vessels with very fine lines and a moderate ratio of length to breadth;

0.440 for vessels with very fine lines and a small ratio of length to breadth;

0.475 for vessels of full forms.

Summing up the wave resistance and skin resistance, determined with the given formulæ, the total resistance may be obtained, and this is always the most appropriate element to rely upon when estimating the I. H. P. of a ship; the result thus obtained may be controlled by the use of I. H. P. constants given in Tables III., IV*a.*, *and* IV*b.*

(To be continued.)

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports under date of January 10, 1905, the following degree of completion of vessels building for the United States Navy:

			Degree of Completion. Per Cent.	
BATTLESHIPS.			Dec. 1.	Jan. 1.
Virginia	19 knots.	Newport News Co.	73.43	75.24
Nebraska	19 "	Moran Brothers Co.	63.64	64.91
Georgia	19 "	Bath Iron Works	69.67	71.94
New Jersey	19 "	Fore River Shipbuilding Co.	72.2	74.3
Rhode Island	19 "	Fore River Shipbuilding Co.	75.1	77.1
Connecticut	18 "	Navy Yard, New York, N. Y.	58.71	61.51
Louisiana	18 "	Newport News Co.	64.73	66.5
Vermont	18 "	Fore River Shipbuilding Co.	29.5	31.5
Kansas	18 "	New York Shipbuilding Co.	35.6	40.1
Minnesota	18 "	Newport News Co.	50.24	52.84
Mississippi	17 "	Wm. Cramp and Sons	15.26	17.84
Idaho	17 "	Wm. Cramp and Sons	13.4	15.39
ARMORED CRUISERS.				
Pennsylvania	22 knots.	William Cramp and Sons	97.03	98.25
West Virginia	22 "	Newport News Co.	97	98.5
California	22 "	Union Iron Works	66.7	68.7
Colorado	22 "	William Cramp and Sons	98.42	99.27
Maryland	22 "	Newport News Co.	93.41	95.14
South Dakota	22 "	Union Iron Works	64.1	65.9
Tennessee	22 "	William Cramp and Sons	58.71	60.53
Washington	22 "	New York Shipbuilding Co	55.3	60.7
SEMI-ARMORED CRUISERS.				
St. Louis	22 knots.	Neafie and Levy Co.	56.6	58.5
Milwaukee	22 "	Union Iron Works	63.2	65.1
Charleston	22 "	Newport News Co.	86.06	88.32
PROTECTED CRUISERS.				
Chattanooga	16½ kts.	Lewis Nixon	99.13	99.13
Galveston	16½ "	Wm. R. Trigg Co.	95.	96.
GUNBOATS.				
Dubuque	12 knots.	Gas Engine and Power Co.	72.3	77.6
Paducah	12 "	Gas Engine and Power Co.	68.4	73.1
TRAINING SHIPS.				
Cumberland	Sails	Navy Yard, Boston	85.	88.
Intrepid	"	Navy Yard, Mare Island	67.5	72.
TRAINING BRIG.				
Boxer	Sails	Navy Yard, Portsmouth	95.	96.
TORPEDO BOATS.				
Stringham	30 knots.	Harlan and Hollingsworth	99.	99.
Goldsborough	30 "	Wolff and Zwicker	99.	99.
Nicholson	26 "	Lewis Nixon	99.	99.
O'Brien	26 "	Lewis Nixon	98.5	99.

THE REPORT OF THE MERCHANT MARINE COMMISSION.

(EXTRACTS.)

A consideration of the serious disadvantages which American ship owners and seamen must meet brings us to the definite, imperative question: What remedy does the Merchant Marine Commission propose to Congress.

Our answer is embodied, as the result of eight months of inquiry and reflection, in the bill "To promote the national defense, to create a force of naval volunteers, to establish American ocean-mail lines to foreign markets, to promote commerce, and to provide revenue from tonnage."

For several years the navy department has been urging Congress to authorize a naval reserve of professional officers and seamen of the merchant service, who shall be in effect a militia of the sea, holding the same relation to the regular navy that is held to the regular army by the organized state militia and national guard.

OUR NEED OF A REAL NAVAL RESERVE.

The United States is the only maritime power, except Russia, which has not made provision for this essential factor of national defense. There is, it is true, a naval militia now existing in a few states, but as a rule it is composed of landsmen and is valuable in war only as an auxiliary in coast and harbor protection, where its advantage is unquestioned. But a naval reserve in the true sense, composed of men habituated to the sea, trained in its difficult work and hardened to its perils, has now virtually no existence in the United States, and nobody understands this better than the good and loyal officers of the state forces, especially those who, because of our sheer lack of such a reserve, were hastily drawn into the naval service on converted yachts or tug boats, or deep-sea cruisers, in the war with Spain. Not a few militia officers and men had at that time their first experience out of sight of land, and for the first time saw the sun go down behind an ocean horizon, "off soundings."

There is, as has been said, one other maritime power besides the United States that is destitute of a trained seafaring reserve, and is in the same plight of fatuous unpreparedness. As a neutral observer, quoted by Rear-Admiral Luce to the commission, said of the Russian Baltic squadron before it sailed away:

> The units of the ships' companies are brave, but as a rule are wanting in the high-sea experience and the elasticity that enable the seamen of the American and British navies to adapt themselves to new and difficult responsibilities as they arise. Nor do the temperaments and traditions of the composite races that man the ships serve to correct these deficiencies. The service is compulsory, not voluntary, and a good portion of the crews is drawn from the interior. The merchant marine is relatively too unimportant to provide a proper nucleus of seamen, and the consequence is that many of the men serving on board ship have no sea aptitude, and are found afloat only because they have drawn an unlucky number in the conscription.

Russia exhausted her best officers and trained men in the first squadron, whose ships now lie sunk at Port Arthur or hide dismantled in the neutral harbors of the Yellow sea. It is not comfortable to think that but for our unbroken victories in the hundred days' war with Spain, Russia's experience with the Baltic squadron might have been our own. It is the evidence of the admiral commanding on the coast of Cuba in that war that the United States had set afloat all its educated officers and skilled seamen in its first battle line, and that if a reverse had come there would have been no crews to man another fleet, even had the new ships been available.

At the present time it is well known that new ships and good ships are being laid up at the navy yards because there are no officers for them, and because the regular navy quota of enlisted men is insufficient for the demands of routine peace service.

THE NAVAL VOLUNTEERS.

The bill as drafted by the commission proposes, as the first essential step in the rehabilitation of our merchant shipping, to create a force of naval volunteers composed of the best officers and men of our merchant ships and deep-sea fishing vessels, and, having created this force, which must necessarily be small at first, to provide means for its healthy and sure expansion. As an inducement to enroll, and in frank recognition of the peculiar national value of a thorough-going seaman, a substantial retainer is offered ranging from $100 a year for the master or chief engineer of a large steamship to $25 for a sailor or fireman, and $15 for a boy. It is understood, of course, that officers and men shall receive regular pay, besides this retainer, during their period of actual naval instruction, and the terms and conditions of this period, which at first cannot be long, and the regulations and qualifications of the service are left, as they ought to be, to be prescribed by the Secretary of the Navy.

Section 2 of the bill authorizes the payment of an annual subvention of $5 per gross registered ton for every vessel, steam or sail, engaged for twelve months in the foreign trade or deep-sea fisheries, $4 for nine months, and $2.50 for six months, provided that the vessel carries among her crew a certain proportion of naval volunteers, and provided further, that the vessel is held at the disposal of the government in war, carries the United States mails, if so required, free of charge, maintains an efficient rating, and makes all ordinary repairs in the United States.

FOR CARGO CARRIERS ESPECIALLY.

To the inevitable question, Will this naval subdivision of $5 per gross ton per annum, payable to a given vessel for no more than ten years, suffice to solve the whole problem of our ocean shipping, creating not only a fleet of capacious and useful cargo ships, but a fleet of fast and luxurious passenger ships?—to this question the commission will frankly reply that no such complete result is to be expected. Regular mail liners of adequate speed on certain important routes are provided for in another way in another section. But this naval subvention of $5 per gross ton does effectually bridge the difference of cost of construction and cost of maintenance, based on wages here and abroad, so far as concerns the average freighting vessel, steam or sail. To this extent it does equalize conditions, and thereby does give our merchant ships a fair fighting chance again upon the ocean.

Take, for example, the actual case of a new steam freighter, a typical cargo vessel, of 3,750 gross tons. Her fixed charges, based on crew wages and maintenance and higher cost of construction, would be, approximately, $13,000 a year greater than those of a British vessel of the same class, with a British crew—if a Chinese or Lascar crew, the difference would be four or five thousand dollars greater. Such a typical American ship would earn a subvention for a year's service, at $5 per gross ton, of $18,750, evening conditions, and perhaps a little more, by way of encouragement to the owners to build new vessels, employ more naval volunteers, and help to make new markets for American commerce.

It is to be noted that one even rate of subvention of $5 per gross ton is provided for all vessels, sail craft included. This is the fairest plan that possibly can be framed. It is simple and intelligible. It is proof against all charges of favoritism and discrimination. Moreover, there is more than one urgent reason why sail vessels should have the same rate as steamers. Our present fleet includes many sail ships of high commercial efficiency. They are often still the pioneers of commerce, visiting new ports where trade is too small or channels too shallow for the steamers. They are the cheapest carriers of certain important cargoes, and their presence is everywhere a check upon exorbitant steam rates.

NOT ENOUGH FOR FAST, HIGH-COST VESSELS.

This proposed rate of $5 per gross ton is unquestionably sufficient to equalize conditions for American sail vessels as against foreign sail vessels, save in the case of the excessively subsidized fleet of France. This same rate, as has also been said, is sufficient to equalize conditions for American cargo steamers, save in exceptional instances. The commission is prepared to anticipate the criticism that this naval subvention will not of itself encourage the building of swift and expensive steamships. Indeed, we are ready to admit this without controversy.

We are frank to say, moreover, that it is our deliberate judgment that in the restoration of the American merchant marine it is the useful, hard-working cargo ship of steam and sail which should have the first and friendliest consideration. For the American people, though they are now sending many compact manufactured goods abroad, are still in the main producers of bulky commodities, so far as concerns their export commerce. Grain, provisions, cotton, lumber, cattle—things like these still make up the greater part of the value, as they do of the volume, of American shipments to foreign countries.

It cannot be too strongly emphasized that the naval subventions offered in Sections 2, 3, and 4 of the proposed bill are not bounties outright or mere commercial subsidies, such as many of our competitors give, but are distinctly based on important public services rendered and to be rendered by the ships and ship owners that receive them. As to the constitutionality and expediency of such guarded subventions as these there can be no question.

NEW OCEAN MAIL ROUTES.

Wherever throughout the country the commission has held its hearings the evidence has been unanimous in favor of the principle of the present ocean mail act, approved March 3, 1891, and a very earnest desire has been expressed for its extension. When this law was passed the rates of compensation originally proposed were cut down about one-third by Congress. Experience has amply proved that this reduction was an error, for, though the older American mail lines have been sustained and developed by the law of 1891 and a few new lines have been created, yet the law has not sufficed to give the United States a complete system of mail communication with the great ports of the world and the chief markets for American merchandise. The American ocean mail lines now operating under contracts provided by the law of 1891, and the compensation received in the fiscal year 1904, are stated by the superintendent of foreign mails as follows:

American Line, New York to Southampton	$690,483.20
Oceanic Line, San Francisco to Australasia	283,203.00
New York and Cuba Mail, New York to Cuba and Mexico	206,082.00
Red D Line, New York to Venezuela and Dutch West Indies	103,325.00
American Mail, Boston and Philadelphia to Jamaica	92,748.00
Total	$1,375,841.20

In the proposed bill not one dollar is added to the expenditure for any one of these five established contract lines. They are left exactly as they are at present, fulfilling, under the law of 1891, their agreements with the government.

Moreover, this law of 1891 is not repealed, and it is not amended except as to certain specific new routes to be established and the new requirements of naval volunteers. Contracts authorized on these new routes are to be made in general in the manner provided for under the existing law, which has stood the test of almost fourteen years of actual experience.

But in one important particular the law of 1891 has undeniably failed. Its reduced compensation has not sufficed to establish contract mail lines to the greater countries of South America, to Central America, to Africa, or to the Orient. Therefore the commission recommends in the proposed bill a substantial increase of compensation on certain specified routes where American steam service will be most likely to increase the foreign markets for American merchandise. These new routes are:

THE NEW SERVICES.

First. From a port of the Atlantic coast of the United States to Brazil, on steamships of the United States of not less than 14 knots speed, for a monthly service at a maximum compensation not exceeding $150,000 a year, or for a fortnightly service at a maximum compensation not exceeding $300,000 a year.

Second. From a port of the Atlantic coast of the United States to Uruguay and Argentina, on steamships of the United States of not less than 14 knots speed, for a monthly service at a maximum compensation not exceeding $187,500 a year, or for a fortnightly service at a maximum compensation not exceeding $375,000 a year.

Third. From a port of the Atlantic coast of the United States to South Africa, on steamships of the United States of not less than 12 knots speed, for a monthly service at a maximum compensation not exceeding $187,500 a year, or for a fortnightly service at a maximum compensation not exceeding $375,000 a year.

Fourth. From a port of the United States on the Gulf of Mexico to Brazil, on steamships of the United States of not less than 12 knots speed, for a monthly service at a maximum compensation not exceeding $137,500 a year, or for a fortnightly service at a maximum compensation not exceeding $275,000 a year.

Fifth. From a port of the United States on the Gulf of Mexico to Cuba, on steamships of the United States of not less than 14 knots speed, for a semi-weekly service at a maximum compensation not exceeding $75,000 a year.

Sixth. From a port of the United States on the Gulf of Mexico to Central America, on steamships of the United States of not less than 12 knots speed, for a weekly service at a maximum compensation not exceeding $75,000 a year.

Seventh. From a port of the United States on the Gulf of Mexico, to Mexico on steamships of the United States of not less than 12 knots speed, for a weekly service at a maximum compensation not exceeding $50,000 a year.

Eighth. From a port of the Pacific coast of the United States, via Hawaii, to Japan, China, and the Philippines, on steamships of the United States of not less than 16 knots speed, for a monthly service at a maximum compensation not exceeding $300,000 a year, or for a fortnightly service at a maximum compensation not exceeding $600,000 a year.

Ninth. From a port of the Pacific coast of the United States to Japan, China, and the Philippines, on steamships of the United States of not less than 13 knots speed, for a monthly service at a maxmium compensation not exceeding $210,000 a year; or for a fortnightly service, at a maximum compensation not exceeding $420,000 a year.

Tenth. From a port on the Pacific coast of the United States to Mexico, Central America, and the Isthmus of Panama, on steamships of the United States of not less than 12 knots speed, for a fortnightly service at a maximum compensation not exceeding $120,000 a year.

Provided, That the requirements of this section as to the rates of speed shall be deemed to be complied with if said rates are developed during a trial of four hours' continuous steaming at sea in ordinary weather in water of sufficient depth to make the test a fair and just one, and if the vessels are maintained in a condition to develop such speed at any time while at sea in ordinary weather. This trial shall be made under the direction and supervision of a board of naval officers which the Secretary of the Navy shall appoint upon the application of the owner or owners of the vessel to be tested.

It will be recognized that in every instance these proposed lines follow natural and important trade routes, and that several of them are valuable, not only for the commercial, but for the political relations which a regular American steamship service will assuredly promote.

NEARLY ALL NEW ROUTES.

As a rule, the ten new ocean mail routes specified in the proposed bill must be created from the beginning—not only the lines, but the ships themselves. There is not one American steamer now running to Brazil, not one to Argentina, not one to South Africa. On the four gulf routes but one American steamer is now found—on the short line to Cuba. In the Pacific ocean the situation is somewhat different. A service from San Francisco to China, Japan, and the Philippines is now maintained by five American steamers of the Pacific Mail and three British steamers of the Occidental and Oriental Company. If the Pacific Mail were to seek the proposed contract, the three British vessels would have to be displaced by new ships of American construction. There is not now, therefore, an "existing line" on this San Francisco route, as the proposed bill contemplates such a service.

Out of the ports of Puget sound only two American steamers of the liner type now run to the Orient and the Philippines. Four other ships of at least equal size and speed would have to be secured to provide a contract service. On this northern route also there is no "existing line," such as would be required by the proposed subvention.

NO "OCEAN GREYHOUNDS."

Frankly, these proposed new mail subventions do not look to the creation of an "ocean greyhound" class. Almost the only "greyhounds" in the world are to be found on a few North Atlantic lines to Europe. It has seemed to the commission that the most useful mail steamships for distant commerce and the mail steamships which the American people most desired at the present time

were modern, efficient vessels, combining moderate speed with large cargo capacity. Such are the steamships called for by the mail subventions of the proposed bill. Their speed, it is believed, is adequate but not excessive. Commercial value is nowhere sacrificed to mere record breaking. At the same time, the stipulated speed is believed to be always at least equal, and in most instances superior, to the average rates of foreign steamers now running in the same or similar services.

For example, only two of all the foreign vessels that last year received United States mails at New York for South America possess even a nominal speed of 14 knots, the rate required from all the new ships of our proposed lines to Brazil and Argentina. Foreign steamers from New York to South Africa are now from 8 to 11 knots, and 10 knots is above the average speed of the foreign craft now trading within the Gulf of Mexico.

On the transpacific routes, the fastest ships now afloat are American built, and belong to the Pacific Mail Company. The required speed for steamers on the North Pacific line (13 knots) is 3 knots less than via the Hawaiian line, but the northern route is the shorter to the Orient, and the winter voyages are often too rough and stormy for high speeding. The three British North Pacific liners are nominally of 16 knots, but it is understood that they are never driven to their maximum.

A NAVAL RESERVE FLEET.

Just as required by existing law, whose safeguards are in no way relaxed, the contracts for the new ocean mail routes of the proposed bill must be awarded, after public advertisement, in free and fair competition, to the lowest responsible bidder offering terms satisfactory to the Post-Office Department, and the ships offered can receive no other subvention or bounty from the United States. As required by existing law, the new mail steamships must be built under naval inspection, and the faster of them must be strengthened to mount powerful guns as armed auxiliary cruisers, while the slower vessels serve as equally indispensable transports or supply ships. The speed of all must be tested on an official trial and certified by the Navy Department, and all these mail ships, of any speed, must be held at the disposal of the government in war. Moreover, they must all carry a quota of men and boys of the naval volunteers. Thus there is guaranteed a new naval reserve of both ships and seamen, and an important reinforcement not only of the commercial power but of the defensive power of the United States.

THE TONNAGE-TAX PROPOSITION.

There now remains to be considered the fourth and last feature of the proposed bill. That is a plan, outlined in Section 8, to increase the tonnage taxes on all vessels, American and foreign, now entering our ports by sea in the foreign trade.

Tonnage taxes are practically the only federal charges levied on shipping in American ports, for entry and clearance fees, etc., are too small to be considered. The present rates of tonnage taxes in the United States are lower than those of the principal maritime nations, and very much lower than the rates of some of those nations. Such charges are sometimes called "light dues," and their original purpose here as abroad was to provide from shipping a fund for lighting the coast for the benefit of shipping. Thus, during the year 1903, the light dues collected in the United Kingdom amounted to £548,196, while the expenses of the British lighthouse establishment were £499,404. From light dues, accordingly, the British government met all cost of lighting the coasts of the United Kingdom, and had a surplus of nearly $250,000 a year to add to an accumulated surplus for other years of nearly $2,000,000.

The commission believes that, with entire propriety, a similar general principle may be adopted in the United States. Our receipts from tonnage taxes in 1903 were $885,841, while the expenses of our lighthouse establishment were $4,538,105. The commission does not propose to raise from tonnage taxes an amount sufficient to meet the entire cost of the lighthouse establishment for several reasons:

First. A relatively small part of our lighthouse establishment expenditures is for the rivers and the Great Lakes. Shipping is here in competition with the railroads, and a federal charge ought not to be imposed on vessels from which railroad traffic is necessarily exempt.

Second. To an extent the same is true of the coasting trade. We have reserved our coasting trade to vessels of the United States, and for more than 20 years it has been the policy of the government to pay out of the public funds many of the charges to which American vessels are subject. Accordingly for many years our coasting trade has been exempt from tonnage taxes. The coasting trade of Great Britain, on the other hand, and of some other foreign nations, is not a reserved trade, being open freely or conditionally to the vessels of all nations. Such countries accordingly, with propriety, levy tonnage or light dues on vessels in the coasting trade.

The approximate cost of the proposed legislation in Sections 1 and 2 for the first fiscal year, from July 1, 1905, to June 30, 1906, may be summarized as follows:

Annual retainers to naval volunteers (3,000)	$150,000
Subventions to 44 registered steamships of 203,871 tons at twelve months' rate of $5 per ton	1,019,355
Subventions to 100,000 tons of square-rigged sailing vessels at nine months' rate of $4 per ton	400,000
Subventions to 100,000 tons of square-rigged sailing vessels at six months' rate of $2.50 per ton	250,000
Subventions to 150,000 gross tons of registered schooners at six months' rate of $2.50 per ton	375,000
Subventions to 50,000 tons of deep-sea fishing vessels at nine months' rate of $4 per ton	200,000
Total	$2,394,355

URGENT NEED OF IMMEDIATE RELIEF.

In the midst of a general condition of buoyant prosperity the American merchant marine in over-seas trade alone of our great national industries is, and long has been, depressed and declining. The slight temporary increase of registered tonnage, due in large part to causes growing out of the Spanish war, has now come to an end, and the absolute cessation of shipbuilding for ocean commerce shows that the country is on the verge of a swift and heavy shrinkage in the small registered fleet still left to it—a fleet actually smaller by 100,000 tons than that of 1810. If there is to be remedial legislation it must be prompt and energetic. Delay only increases the cost and intensifies the difficulty of the undertaking.

It may be said without exaggeration that there is not a large ocean shipyard in America, and not an ocean steamship company, except the few mail lines operating under the act of 1891, that is not looking to instant and vigorous remedial legislation by congress as the one hope of its continued existence. Within a few weeks an important shipyard on the Delaware river, after a long and brave fight against adversity, has succumbed and gone into the hands of receivers. This yard has a splendid modern plant, zealous and capable managers, and the prestige of an active career of half a century. The American government and people may well ask themselves this grave question: Where in a few years can they find solvent shipyards to contract with to build their battleships and cruisers unless the complete paralysis now threatening this great industry is speedily arrested by national laws?

If the passage of the legislation proposed by the commission is postponed to the next session of Congress a condition already desperate will have become still more desperate. The time to act is now. The commission has prepared a conservative measure, aiming to achieve its purpose at a minimum cost, fair to all sections and interests, and directed especially to the strengthening of the national defense and the extension of American commerce to new and distant markets. The commission can see no reason why a cautious measure of this kind, making no large immediate draft upon the national revenues, cannot be passed at the present session of Congress.

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Notice to Advertisers.

Changes to be made in copy, or in orders for advertisements, must be in our hands not later than the 15th of the month, to insure the carrying out of such instructions in the issue of the month following.

The American Merchant Marine.

We wish to call attention to the extracts, in another column of this issue, from the report made to the Senate by Senator Jacob H. Gallinger, Chairman of the Merchant Marine Commission. The condition of our oversea shipping is so deplorable, and the matter so urgent and of such intimate interest to maritime men generally, that it is sincerely to be hoped that something will be done during the present session of Congress to provide relief. The dispatches in the daily papers would seem to indicate that there is a strong probability that this matter may be side-tracked, and allowed to wait another year before anything is done. It seems almost incredible, when no one disputes the urgency and supreme importance of the case, not only from a sentimental, or even from a purely commercial point of view, but from a consideration of the possible requirements for the national defense, that our legislators in Washington can allow such a measure to go by default. It cannot be the cost of operation of the scheme as proposed, at which they balk; for that will be very largely met by the new tonnage tax, which forms an integral part of the proposition. Party considerations certainly cannot dictate any hold-up policy, for both parties are pledged to do what they can for the languishing industry.

This is not the place for any discussion of the subject from a political point of view, even if, as in the present instance, a subject on which the parties are agreed may be said to have a political aspect; but so much of the future of American shipping and shipbuilding depends upon this very thing, so much of the weal or woe of the nation may well be bound up in this, the only neglected branch of its field of activities, that we feel called upon to add what we can to the demand for a prompt and satisfactory solution of the difficulty on the part of Congress.

Were no other argument available, the present maritime plight of Russia, referred to in the report, and one due largely to causes underlying in part our own marine impotency—causes which the operation of the new bill would do much to alleviate, if not altogether to annul—ought to point the moral, and exercise a potent influence in favor of immediate action. But the case does not appeal alone to the fear of national humiliation, remote though that contingency may seem. National pride and a sense of justice are likewise bound up in the matter, for it is not just to ourselves that we continue supinely to allow the greatest carrying trade of the world, a trade which is ours by all the laws of logic, to be taken away from us by other powers, while our own coffers are drained to pay their bills for the service. It has been estimated that the transportation of our foreign commerce involves the expenditure of no less than two hundred millions of dollars per annum. No one will pretend to argue that anything less than fifty percent of this carriage is a fair proportion for us; and this saving of one hundred millions of dollars every year—for, from the point of view of the nation as a whole, it is a saving—would do an immense service in improving conditions among the seafaring and shipbuilding population in this country, would operate to augment their numbers, and place the industry upon that firm basis which characterizes the position of perhaps every other legitimate industry in the United States. Though Abraham Lincoln was arguing on another subject, we cannot but recall, in connection with the question of the payment of these enormous freight bills to foreigners, his remarks about Mrs. Lincoln's cloak: "If she buys it abroad, we get the cloak, and the foreigner gets the money; but if she buys it in this country, we have the cloak, and the money too."

The necessity of the case being conceded, the only question that remains is as to the means to be employed for accomplishing the results desired. The views of the majority are sufficiently well indicated in the extracts to which we have called attention, and provide substantially for a certain subvention, together with provision for enrolling certain parts of the crews of American ships in a corps of naval volunteers, with suitable retainers for the various grades of officers and men. These two features, together with the provision for increased tonnage taxes, constitute the main points of the scheme as proposed by the majority of the Merchant Marine Commission.

The Minority Report, presented to the Senate by Mr. Mallory, gets at the subject in an entirely different way. Arguing that the subvention is nothing more nor less than a new form of the subsidy advocated by the late Senator Hanna and repudiated by Congress some time ago, a plea is made for the establishment of a system of preferential or discriminating duties, and arguments are advanced showing how in the early days of our nation's history such a system was put into effect with splendid results. It is stated that one of the first acts of the first Congress was to establish a tonnage tax of six cents per ton on American vessels and fifty cents on foreign vessels in our ports, and to provide for a rebate of ten percent on the tariff duty charged against goods imported in American vessels. It is further stated that the result of this system was that our mercantile marine advanced by enormous strides, from the carrying of seventeen percent of our imports and thirty percent of our exports in 1789, to ninety-one percent of the imports and eighty-six percent of the exports five years later.

Against this feature of the Minority Report it is argued that a large number of trade conventions with foreign nations must be abrogated, and that we would then face the probability of retaliation against our shipping in foreign ports, and more particularly against our exports, especially food stuffs. It does not appear, however, that there is much danger of this latter form of retaliation, because the only effect would be to increase to the foreign consumer the cost of the necessities of life, and it is hardly probable that any foreign nation would undertake this sort of discrimination. There is one plan of discriminating duties, however, which has been quite prominently advocated, and which involves no possibility of retaliation, and that is a duty discriminating against foreign ships which carry products to the United States from countries other than their own. This provision, of course, would be aimed almost entirely at German, English and Norwegian shipping, and would place in American hands to a large extent the carriage of goods from such nations as have only small merchant marines.

Whatever may be the result of the deliberations of Congress in connection with these two reports, we certainly hope that something will be done immediately, which will have the result at which both sides of the question aim—the resumption by the American flag of the high position which it once occupied in the ocean commerce of the world.

Our Enlarged Page.

We have preached standardization to such an extent that nothing has remained for us but to adopt this policy by enlarging Marine Engineering to "standard size," nine by twelve inches in measurement. A more important reason for the enlargement was the fact that the smaller page did not give us the scope that was necessary in illustrating many articles.

The Screw Propeller.

Of the numerous items making up the design, construction and operation of a steamship of the present day, there is none the importance of which much transcends that of the screw propeller; yet there is certainly none which is less well understood by those who have in charge either its design or its operation. The propeller of the present day is very largely an evolution. This evolution has proceeded, not along strictly scientific lines, but, if we may so state it, along the lines of least resistance. One designer has followed blindly in the footsteps of another, giving vent perhaps to a few of his own ideas in the matter, but being in the main fully as mindful of precedent and subservient to it as would cheer the heart of the most pettifogging lawyer to be found in a day's journey. The inevitable result of such a condition of affairs has been to leave the science of the design of screw propellers, as viewed from the side of practice, in about as chaotic a state as could well be imagined. For fulfilling the same set of conditions two designers will produce two propellers of totally different characteristics, and yet, strangely enough, both will be about equally suited to the task in hand.

The existence of such conditions would appear to indicate that there is no scientific basis for the design of the screw propeller, which conclusion is by no means in accordance with the truth. A certain amount of experimentation has given rise to a knowledge of certain facts connecting the design of the propeller with its operation, and these experiments are being extended at the present time, in the hope of developing a further relation between theory and practice. The theory which has appeared to date, however, has been in general so abstruse as to defy all attempts of the practical man to make satisfactory use of it. Appreciating this condition of affairs, Professor W. F. Durand has prepared for us a series of articles on the subject, in which, as he stated in the opening paragraph of the first of the series, published last month, "an effort will be made to treat the screw propeller in a simple and practical manner with reference to form and structure, delineation on the drawing board, design and adaptation to specific purposes, relation between conditions of operation and results obtained, and features relating to operation under special conditions. The purpose of these articles is to provide a simple answer, in so far as may be possible, to many of the practical questions which are continuously arising in connection with the design, construction, and operation of screw propellers for marine propulsion."

We desire to call especial attention to these articles, which will be continued next month by the question of "Design." Shorn of all the complex mathematical embodiment of the usual treatment of the subject, they should prove of great interest and practical benefit to all who are connected with either the design or the operation of the screw propeller.

ABSTRACTS FROM RULES AND REGULATIONS OF LLOYD'S REGISTER OF SHIPPING.

EDITION OF 1903.

ARRANGED BY HARRISON S. TAFT.

PART II.

25. Riveting.

1. With plates of two thickness, the size of rivet to be governed by the heavier plate.

2. All double and treble riveting should be chain fashion, except for keels, stems, and stern posts, which should be zigzag.

3. Bar Keels, Stems, and Stern Posts to be double riveted. Rivets to be 1-4 inch larger in diameter than otherwise called for, but not to be over 1 1-4 inch diameter. In single screw vessels above 350 feet in length the part of stern frame below boss to be treble riveted to shell.

4. Flat Keel Plates, Floors, Web Frames, Vertical Keels and Keelsons, Rider Plates to be treble riveted throughout.

5. Laps and edges of Bulkheads and Deck Plating to be single riveted. Laps or edge butts of bulkhead plating to floors to be double riveted.

6. All Shell Butts, Deck Stringers, Deck Plate Butts for 1-2 length amidships, Deck Tie Plates, Tank Top Margin and Middle Line Strake of Double Bottom Vessels, and Plate Stringer, to be at least double riveted in all ships. See No. 11 below.

TABLE 12.—THICKNESS OF DOUBLE BUTT STRAPS FOR SHELL PLATING, SHEER STRAKE AND DECK STRINGERS.

Thickness of Plate in 20th	9	10	11	12	13	14	15	16	17	18	19	20	21	22
Straps for cks Rivets in 20th.....	7	8	8	9	10	10	11	12	12	13	14	14	15	15
Straps on opposite Side in 20th...	6	6	7	7	8	9	9	10	11	11	12	13	13	14

TABLE 13.—BUTT STRAPS AND RIVETING SCHEDULE FOR SHELL PLATING AND DECK STRINGERS.

Second Numbers.	Not over 8,000.	[1]Over 8,000. Not over 13,000.	[1] Over 13,000 Not over 16 000.	[1] Over 16,000. Not over 20,000.	[2] Over 20,000. Not over 24,000.	[2] Over 24,000. Not over 28,000.	[2] Over 28,000.
Butt straps of :							
Sheer strake.....	D R+$\frac{1}{20}''$	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{3}{20}''-\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+$\frac{4}{20}''-\frac{1}{2}$ L	T R T+$\frac{4}{20}''-\frac{3}{4}$ L
First bilge strake.....	D R T+$\frac{1}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{3}{20}''-\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+$\frac{4}{20}''-\frac{1}{2}$ L	T R T+$\frac{4}{20}''-\frac{3}{4}$ L
Second " "	D R	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{3}{20}''-\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+$\frac{4}{20}''-\frac{1}{2}$ L	T R T+$\frac{4}{20}''-\frac{3}{4}$ L
Third " "	D R	D R	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{3}{20}''-\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+$\frac{4}{20}''-\frac{1}{2}$ L	T R T+$\frac{4}{20}''-\frac{3}{4}$ L
Other outside strakes...	D R	D R	D R	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{3}{20}''-\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+$\frac{4}{20}''-\frac{1}{2}$ L	T R T+$\frac{4}{20}''-\frac{3}{4}$ L
" inside "	D R	D R	D R	D R	T R+$\frac{3}{20}''-\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+$\frac{4}{20}''-\frac{1}{2}$ L	T R T+$\frac{4}{20}''-\frac{3}{4}$ L
U D stringer.....	D R+$\frac{1}{20}''$	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R+$\frac{2}{20}''-\frac{1}{2}$ L	T R Dbl−$\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+Dbl−$\frac{1}{2}$ L	T R T+Dbl−$\frac{1}{2}$ L
M D "	....	....	....	....	T R+$\frac{3}{20}''-\frac{1}{2}$ L	T R−$\frac{3}{4}$ L+$\frac{4}{20}''-\frac{1}{2}$ L	T R T+$\frac{4}{20}''-\frac{3}{4}$ L
End straps.....	D R	D R	D R	D R	D R+$\frac{2}{20}''$	D R+$\frac{2}{20}''$	T R+$\frac{2}{20}''$
Wide shell plates.....	T R+$\frac{1}{20}''$	T R+$\frac{1}{20}''$	T R+$\frac{1}{20}''$	T R+$\frac{1}{20}''$	T R+$\frac{1}{20}''$	T R+$\frac{4}{20}''$	T R+$\frac{4}{20}''$

Flat keel plates are to be triple riveted throughout.
[1] Triple riveted butts may omit alternate rivets in back row.
[2] Triple riveted butts to have back row of rivet spaced 5 to 5½ diameters. All triple riveted lapped butts to have three complete rows.
D R = double riveted; T R = triple riveted; T R T = triple riveted throughout, D R T = double riveted throughout.
Plus sign (+) means *with*. Minus sign (−) means *for*.
Examples: T R + $\frac{2}{20}'' - \frac{1}{2}$ L means: triple-riveted butts with butt straps $\frac{2}{20}$ of an inch heavier than plating for ½ length amidships.
T R − ¾ L + $\frac{4}{20}'' - \frac{1}{2}$ L means: triple-riveted butts for ¾ length amidships with butt straps $\frac{4}{20}$ of an inch heavier for ½ length amidships.

7. Longitudinal Laps of Shell Plating to be single riveted for bottom plating, from keel to upper turn of bilge, of less than 7-20 inch thick, and for side plating, from upper turn of bilge to gunwale bars, of less than 9-20 inch thickness. Otherwise, laps are to be double riveted. Lower edge of sheer strakes to be double riveted in all cases. The style of midship riveting to be carried to ends of ship. When length of ship is 480 feet or above, the longitudinal laps are to be treble riveted for 1-3 depth of vessel in fore-aft part of ship for a distance of 1-4 vessel length. When of 450 feet to 480 feet long to have said treble riveting as called for.

8. Lap Butts of Shell Plating for vessels whose second number is under 16,000, to be treble riveted for 1-2 length amidship and double riveted at ends. When said number is or above 16,000 said butts to be treble riveted throughout; with three complete rows in all cases.

9. Shell Straps and Deck Stringers.

a. The boss plate butts to be treble riveted when second number is 18,700 or above, as per specified.

b. Outside shell strakes over 40 inches and not exceeding 46 inches in width; inside shell strakes 48 inches and not over 54 inches in width, to have treble riveted butts.

10. Single riveted Shell Seams to have one frame rivet through landing edge at each frame. Double riveted seams to have one rivet only, except in cases of joggling, when two rivets must be used. Treble riveted seams to have two such rivets (the middle one being omitted.)

11. Double Bottom.

a. Butts and edges of middle strake plating to be in all cases double riveted. In engine and boiler room all butts and edges of plating to be double riveted. Butts of side girders and margin plates to be double riveted. When second number is under 21,000 the center line girder butts are to have double straps double riveted. When said number is over 21,000 the butt of center, side girders, and margin plates to be triple riveted with alternate rivet in back now left out.

b. When second number is 20,000 and under 30,000, all butts of inner bottom plating to be double riveted for 1-2 length amidship.

c. When said number is 30,000 and under 38,000, all butts of inner bottom plating and an additional strake edge on each side of middle line strake are to be double riveted throughout.

d. When said number is 38,000 and under 51,000, the remaining edges to be double riveted for 1-2 length amidship.

e. When longitudinals are worked on top of floors, all butts (except of margin plate) and edges of all plating may be single riveted.

12. *a.* Lower Masts. Seams to be double riveted. In masts of less than 84-foot length they may be single riveted if angle stiffeners are used. Butts below partners to be double, those above triple riveted. Seams of steamers' masts with fore-aft sails, or of masts used for auxiliary purposes, may be single riveted throughout.

b. Topmasts and Yards to have single riveted seams and treble riveted butts.

c. Bowsprits to have double riveted butts inside of wedging, trbele riveted outside of wedging.

13. Awing Deck stringers and side plating to have double riveted butts.

14. Rudder Rivets to equal those in upper edge of garboards amidships for side plate rudders, and to those of shell to stern frame for single plate rudders.

15. Doubling Plates should be riveted up in the center of plate before at edges.

16. There should be at least four rivets in each flange of stringer shell clips when frame spacing is less than 29 inches; otherwise 5 rivets, up to a 32-inch frame spacing.

17. When fore-aft flange of frame does not exceed 3 inches, shell rivets to frames should not exceed 7-8 inch diameter; when said flange is 3 1-2 inches, not over 1-inch rivets are to be used.

18. Angle bosom pieces to take 3 rivets on each side of butts. Those for reverse bars may have but 2 rivets, except for a butt on center line, where 3 rivets must be used.

19. Floor clips to have 3 rivets to floors.

20. Deep frame angles to have single riveted lap to frame.

21. Side stringer plate brackets to have two rows of rivets to said stringer at bulkheads.

22. Shell butts with a single treble riveted strap, when second number is 20,000 and under can have alternate rivets of back row left out. When above 20,000, said rivets may be spaced 5 to 5 1-4 inches apart. All lapped butts to have 3 complete rows.

23. Countersunk. Depth to equal thickness of plate when less than 14-20 inch, but 9-10 of said thickness when plate is or above 14-20 inch thick.

26. Double Bottoms.

1. When frames are cut at margin plates the margin plate bottom and top frame brackets to be worked at every frame.

2. Longitudinal Girders Places on Top of the Floors should not exceed 3 feet apart. They are to have a continuous top and bottom angle. Margin plated top frame brackets to be worked at every frame. When reverse bars are cut short doubling pieces should be worked to frames in way of margin plate.

TABLE 14.—DOUBLING OF CLIPS ON MARGIN PLATE TOP FRAME BRACKETS, AND SPACING OF GUSSET PLATES.

Second Number.	Extent of Double Clips on Margin Brackets.	Gusset Plates.
*20,000 under 40,000....	½ L amid.	....
30,000 " 40,000....	⅝ L "	Every 4th frame.
40,000 " 50,000....	†¾ L "	" 3d "
50,000 and up.....	Throughout.	" 2d "

*Deep or web frame vessels only.
† Measured from collision bulkhead.
Single clips on outside of margin plate should be worked opposite to floor or bracket clips on underside of margin plate.

3. Cellular Bottoms with Continuous Longitudinals.

a. When second number is or under 11,000, triangular bracket plates are to be fitted on each side of longitudinals at top and bottom on alternate frames fore-aft of machinery space. When said number is 11,000, solid man-hole floor plates are to be fitted to alternate frames throughout double bottom.

b. Center keelson brackets are to be fitted to the intermediate frames on all ships with flat keels, and in vessels with hanging keels when second number is 18,000 or above. When second number is 38,000 and under 51,000, said brackets and those at margin plate to be wide enough on top to take 3 rivets of an intermediate reverse angle for 3-5 length. Floor plates should be secured to center vertical keelson with single clips unless second number is 24,000 or above, when they should be doubled for 1-2 length amidship. Inside and outside margin plate frame brackets to be fitted to every frame.

c. Floors should be fitted on every frame between collision bulkhead and the 3-5 length of vessel amidship.

4. Cellular Bottoms with Solid Continuous Floors.

a. The floors are to be worked in one piece from center line to margin plate and placed on every frame, with double clips to center vertical keelson for 1-2 length amidship when second number is 24,000 or above. Intercostal longitudinal girder to be worked at about 1-2 distance from center line to margin plate, and they are to be well clipped to shell and inner bottom.

b. When the vessel's breadth from margin plate to margin plate is 34 feet and not over 44 feet, two intercostal longitudinal girders must be used.

5. Provisions should be made for a thorough ventilation of inner bottom spaces by cutting liners short or otherwise.

6. Solid man-hole floors are to be fitted to every frame in engine room and to alternate frames in boiler rooms with double top angles and clips throughout.

7. When double bottom is worked throughout boiler and engine spaces a well should be provided at aftermost frame space in engine room.

8. The inner bottom plating to be worked fore-aft and to be covered with at least 2 1-2 inches of wood.

9. Intermediate reverse stiffening bars should be worked to inner bottom plating unless longitudinals are fitted closer than specified with solid floors fitted to alternate frames.

10. Double bottom ships built under Nos. 2 or 4 can have shell plating in way of same (except keel and garboards) reduced by 1-20 inch when said plating is 11-20 inch or above.

11. If flange bracket plates are used they should be 1-20th heavier than otherwise.

27. Hatch Ways and Deck Openings.

1. Minimum heights of hatch way coamings above weather decks should be as follows:

On shelter, bridge, or awning decks, 18 inches.

On upper, spar, or raised quarter decks, 24 inches.

On upper decks in wells of well-deck vessels, 30 inches.

2. Hatch coaming deck angle to extend 1-2 inch above wood deck. Alternate spaced half beams to be secured to coaming plate with double clip; every frame spaced beams with single clip. Three-rivet connection to coaming plates for half beams of 7 1-2 to 9 inches in depth; 4 rivets for depth of 10 to 12 inches.

3. Web beams to have double angles on top and bottom and to extend to lower edge of fore-aft coaming plates. To equal the thickness of end coamings. Center depth of web plate for shallow coamings on middle deck to be 1 1-4 end depth and ends are to extend to lower edge of coaming.

4. Hatch covers should be from 2 1-2 to 3 inches thick, with a 1 3-4-inch bearing at edges. Cleats for holding down tarpaulins to be spaced not more than 2 feet apart.

5. All coaming plates to extend to lower edge of beams at end of hatches.

TABLE 15.—HATCH COAMINGS.

Weather Deck.			Middle and Lower Decks.			
Weight of Coaming Plates.			Depth and Weight of Coaming Plates.			
Hatch Length in Ft.	Side.	End.	Hatch Length in Ft.	[1] Depth.	[2] Side.	End.
Under 12	$\frac{7}{20}$″	$\frac{7}{20}$″	10 under 14	16″	....	....
12 " 16	$\frac{7}{20}$″	$\frac{7}{20}$″	14 " 18	18″	....	....
16 to 24	$\frac{7}{20}$″	$\frac{7}{20}$″	18 to 24	20″		....

[1] Depth from under side of carling to top of coaming.
[2] $\frac{1}{20}$″ thicker than for an upper deck hatch side coaming of same length.

TABLE 16.—HATCH BEAMS.

Hatch Beams for Deck Hatches of 12′ and under 16′ in Length.		Number and size of Fore-Afters Resting Upon Web Beams.			
Width in Feet.	Number of Beams.	Hatch Width.	No.	[1] Bulb Plate. Center.	Side.
Under 12′	One 8″ beam	6′ under 8′	1	7″ × $\frac{7}{20}$″	....
12′ " 16′	" 9″ "	8′ " 10′	1	8″ × $\frac{8}{20}$″	....
Hatch beams for deck hatches, 16′ to 20′ in length		10′ " 12′	3	8″ × $\frac{8}{20}$″	5″ × $\frac{5}{20}$″
Length in Feet.	Number of Beams.	12′ " 14′	3	9″ × $\frac{9}{20}$″	6″ × $\frac{6}{20}$″
16′ to 20′	One web plate	14′ " 16′	3	10″ × $\frac{10}{20}$″	7″ × $\frac{7}{20}$″
20′ " 24′	Two " plates	.. ..	..	....	....

[1] Wooden ones may be used if of equal strength.

28. Deck Houses. Boiler and Engine Casings.

1. Engine or boiler room skylights which are located on a poop or bridge house deck, should have coamings at least 2 feet high.

2. Engine and boiler deck houses on a weather deck are to have coamings at least 18 inches above said deck, with steel casing to about 7 feet above the deck. Thickness of plating to equal that of side plating of a poop, with coaming plate 1-16 inch heavier than for trunk bulkheads. In awning-deck vessels said steel casing need not exceed 4 1-2 feet above deck if a skylight is worked on top with iron covers and a 9-inch or more coaming. Said deck house and between decks casings to have stiffeners spaced 30 inches apart equal to reverse bars. Doors in said deck houses and said casings to have lower edge 18 inches above deck.

29. Masts, Yards, Bowsprits and Chain Plates.

1. All masts of 75 feet or more in length (from cap to top of keelson) to be built of 3 strakes, and plates should be doubled in way of partners. Three lines of stiffeners extending entire length of mast are to be worked on inside when masts are 84 feet or above from cap to top of keelson. Diameter of masts at partners is to be 1-3 of length as specified above. Diameter at heel to be 3-4 of diameter at partners + 1-16 inch. Diameter of head to be 5-8 diameter at partners + 1-16 inch.

2. Bowsprits over 28 inches in diameter to have a vertical web plate from within the wedging to gammoning, with single angle bar connection to plating, and with two additional angle bar stiffeners. When diameter is 28 inches or below, to have angle bar at center of plate for whole length of sprit. Plates to be doubled at wedging.

3. TOPMASTS to be doubled in way of lower mast cap. When 38 feet in length and under 46 feet, to have two angle stiffeners 3 inches by 2 1-2 inches by 6-16 inch on fore and aft parts of masts. When length is 46 feet and under 66-foot angles to be 3 1-2 inches by 3 1-2 inches by 6-16 inch. Diameter at lower cap, sheave hole, and topmast cap to be not less than that of yards at the places.

4. YARDS to be doubled at center, extending beyond truss hoops.

5. STEAMERS. Lower and upper masts and bowsprit fitted for auxiliary purposes only, may be 1-8 less in diameter than otherwise. When said masts are to carry fore-aft sails only, they may be 1-5 less in diameter than otherwise.

6. Chain plates to be in proportion to vessel's size and be well secured to sheer strake when possible.

30. Sluice Valves, Ventilators, and Freeing Ports.

1. Ventilator coamings should stand at least 3 feet above a weather deck. When under 12 inches in diameter to be of 1-4-inch plate; when of 12 inches and under 15 inches diameter 5-16-inch plate; when of 15 inches to 24 inches diameter 3-8-inch plate.

2. Doubling plates to be fitted under all sounding pipes.

3. Sluice or gate valves must not be fitted to forward collision bulkhead. If fitted to other water-tight bulkheads they must at all times be accessible, with rods extending above load water line.

When aft peak is used as a ballast tank no sluice valve or cock is to be fitted to aft collision bulkhead. But if used otherwise and no pump is connected to it, a sluice valve may be placed on said bulkhead.

4. Soil pipes discharging below load water line must have an iron pipe for some distance above load water line from opening. If the upper part of the pipe be of lead it must be properly protected with either zinc or iron.

5. All head and stern pumps must be fitted with sea-cocks to the outside plating and shall always be accessible.

TABLE 17.—AREA OF FREEING PORTS.

Length of well bulwarks in feet.	30	35	40	45	50	55	60
Total area, each side, in square feet...	9.5	10	10.5	11	11.5	12	12.5

65 feet and above: 1 square foot for each 5 feet of length. Ports to have yellow metal pins in hinges and are to swing outboard.

31. Wood Decking, etc.

1. When laid on an upper steel deck to be 3 inches thick for pine and 2 1-2 inches for teak. When laid on a middle steel deck 2 1-2 inches thick.

2. When a 4-inch deck is worn to 3 inches, a 3 1-2-inch to 2 3-4 inches, and a 3-inch to 2 1-2 inches, they should be renewed.

3. Cargo batterns to be worked in holds and all enclosed cargo spaces.

4. Deck planks 6 inches or less in width to have single fastenings; when over 6 inches double fastenings. Upper deck to be fastened with galvanized screw bolts, with nuts at under side of the beam flange and tie plates. Deck plans to be scored over diagonal tie plates.

32. Steering Gears and Rudders.

1. Diameter of steamer's rudder to be determined by the formula:

$$d = \tfrac{1}{82}\sqrt[3]{D \times b\,(2B - b)S^2}.$$

Where D = draft in feet, B = greatest distance in inches from center of pintles to back of rudder; b = greatest breadth of rudder in inches, and S speed in knots.

2. Diameter of chains in inches is found as follows:

$$d = .38\sqrt{\frac{D^3}{r}}.$$

Where D = diameter of rudder head in inches, and r = radius of quadrant in inches.

3. Rods to be 1-4 larger in diameter than chains as above.

4. Diameter of sheaves to be at least 16 diameters of chain; pins are to be twice diameter of chain.

5. Steering engine stops should be set at a slightly less angle than rudder stop. Rudder side plates to equal lower part of bulkhead plating.

6. Springs to be attached to all steering gear of steamers.

7. Rudder pintles to be spaced 4 to 5 1-2 feet apart with top pintle as near the rudder's head as possible.

8. Steamers above 250 feet in length to have both steam and hand steering gears.

9. Spare tiller and gear to be provided with ships that do not have two independent steering gears.

10. Cast steel rudders, quadrants, and tillers of ordinary design to be dropped onto hard ground from a height of 7 to 10 feet and then to be well hammered with at least a 7-pound hammer.

33. Boat and Anchor Davits.

Diameter of boat davits in inches to be 1-5 length of boat in feet when of usual proportions. Otherwise:

$$\sqrt[3]{\frac{L \times B \times D}{40}\left(\frac{H}{3} + S\right)}.$$

Where L, B, D, are the boat's dimensions, H = heights of davit above top support, and S = spread, all expressed in feet.

TABLE 18.—DIAMETER OF ANCHOR DAVITS.

Weight.	Spread in Feet. 9	10	11	12	13	14	15
2,240	6″	6¼″	6½″	6¾″	6¾″	7″	7¼″
2,800	6½	6¾	7	7⅛	7¼	7⅛	7¾
3,360	7	7¼	7½	7¾	7¾	8	8¼
3,920	7¼	7½	7¾	8	8	8⅛	8½
4,480	7½	7¾	8	8¼	8½	8¾	9
5,040	8	8¼	8¼	8¾	9	9⅛	9½
5,600	8¼	8½	8¾	9	9¼	9½	9¾
6,160	8½	8¾	9	9¼	9½	9¾	10
6,720	8¾	9	9¼	9½	9¾	10	10¼

TIE RODS.

Diameter of main post at deck	6″	6½″	7″	7½″	8″	8½″	9″	9½″	10″	10½″
" " tie rod	1¾	1⅞	2	2⅛	2¼	2⅜	2½	2⅝	2¾	2⅞
" " jib (middle)	3	3¼	3½	3¾	4	4¼	4½	4¾	5	5¼

With two tie rods, each to be three-quarters diameter of single rods.
When anchor chains are worn to four-fifths of their original cross-section area, said secion should be renewed.

34. Three-Deck Vessels.

1. Three-deck vessels to be at least 17 feet to middle deck with two or more complete decks and calked and with hold beams or web framing, etc.

2. Sheer strake to be at upper deck gunwale and middle deck strake to be an outside one. Sheer strake, upper and middle deck stringers, and three bilge strakes to be treble riveted for 1-2 length. (See Table 13.)

35. Spar-Deck Vessels.

1. Said vessels to have three tiers of deck beams and be not less than 17 feet in depth to main deck. Being of a lighter construction than three-deckers, they are to have two sheer strakes; one at spar deck the other at middle deck. Both sheer strakes, spar, and middle deck stringers, and three bilge strakes to be treble riveted for 1-2 length. Main deck sheer strake to have its lower edge not more than 1-2 its depth below main deck stringer. Wood decking 3 1-2 inches thick to be fitted on main deck.

36. Awning Deck Vessels.

1. This class consists of those having a light upper structure over the main deck proper, with rounded gunwale, beams are to lap frame 18 inches. Awning deck beams to be on alternate frames. Engine and boiler enclosures as for ordinary weather deck.

37. Poop, Forecastle, Bridge-House Vessels.

1. In way of these superstructures a reduction of 1-4 from the dimensions, which would be necessary if the vessel was flush deck, will be allowed in outside plating stringer and tie plates upon beams, angle bars, or stringer plates and flat of deck.

2. Beams of these decks to be spaced on alternate frames with an 18-inch lap to frames if rounded form of gunwale is used. Side plating in way of poop, forecastle and bridge houses above sheer strake need not exceed that for awning vessels, and have double riveted butts.

3. When poop exceeds 1-4 vessel's length the sheer strake is to be worked double and upper deck stringer increased in way of break for 20 to 30 feet.

4. When combined length of a poop and bridge-house deck, or of a raised quarter deck and bridge-house deck exceeds 2-5 length of vessel, and second number is or above 15,000, the sheer strake is to be doubled for 1-2 length amidships. In vessels whose bridge house is not less than 2-5 length the aforesaid doubling may be dispensed with if other specified strengthening is used. Bulwarks at ends of the bridge house to be increased in weight.

38. Raised Quarter Decks and Sunken Forecastles.

1. Number and arrangement of hold beams, beam stringers, and hold stringers in way of a raised quarter deck, and the height of reverse bars on frames must be in accordance with the rules for the increased depth of vessel.

2. Main sheer strakes to be doubled at break unless raised quarter deck connects with bridge house, in which case side plating of the raised quarter deck is to be doubled. If the side plating of raised quarter deck is not doubled it must be increased and sheer strake doubled at said point. Bulwark plate of raised quarter deck adjoining the bridge side plating to be increased in weight; the sheer strakes to be doubled at front of bridge. Butts of side plating of raised quarter deck, of main sheer strake and of strake next below, to be treble riveted at break with heavier straps. Main deck stringer to extend abaft of brake about seven frame spaces and the raised quarter deck stringer forward about four frame spaces. Bridge stringer also to extend aft of brake. Raised quarter deck to be connected to deck bridge house with double angles.

3. *a.* With vessels of extreme proportions requiring a steel deck to be worked on main deck said deck plating should scarph raised quarter deck for two to five frame spaces and have four to five diaphram plates with double clip connections to decks and brake bulkhead, with an angle on aft face, with doubling of raised quarter deck side plate at said point for 18 to 20 feet in length; web plates of at least 15 inches wide are to be fitted to forward side of brake bulkhead in way of diaphragm plates, with brackets to decks at top and bottom.

b. When second number is 24,000 and under 26,000, scarphs to cover four frame spaces with 18-inch webs on

bulkhead, with doubling of raised quarter deck side plates from 1-4 length from stern to 8 feet beyond brake bulkhead. When said number is or over 26,000, to have a five-frame scarph with web and doubling of sides as herein stated.

c. When second number is over 20,000 or ratio of length to depth exceeds 13, the brake bulkhead to be at least four frame spaces abaft aft end of engine room opening.

4. When vessels are of ordinary proportions as not to require the above scarphing of main and raised quarter decks, four or five bracket knees should be fitted on both sides of break bulkhead.

5. When combined length of a poop and bridge house, or of a raised quarter deck and bridge house exceeds 2-5 vessel's length, and second number is or above 15,000, the sheer strake is to be doubled for 1-2 length amidships.

6. When depth to top of a raised quarter deck beam is 24 feet or over and lower deck hatches are not framed, a web frame must be worked abreast of hatches, extending from floor to upper deck.

39. Rivet Spacing, Lloyd's Register.

1. Edge laps, center to center of rows, not less than ..... 1 1-2 diameters.
2. Butt straps, " " " " " " " " ..... 2 "
3. Butt laps, " " " " " " " " ..... 2 1-2 "
4. *a.* Butts of shell plating; butts of upper spar and middle deck stringers, and of stringers of bridge decks exceeding 1-3 length of vessel amidships (except quadruple riveted butt laps) 3 1-2 "
b. With single butt strap treble riveted, when second number is under 20,000, alternate rivet in back row may be left out. When second number is 20,000 or over, back row to be spaced 5 to 5 1-4 "
(All lapped butts to have three complete rows.)
5. Quadruple riveted butt laps; butts of deck plating; of margin plates; of girders; of lower deck and hold stringers; of tie plates; of floor plates; of stringer plates on other deck erections; butts and edges of inner bottom plating; edges of shell plating (forward and aft) for a frame spacing of 26 inches or over; landing edges of plating in flat of bottom before 3-5 length in vessels of full form forward..... 4 "
6. Edges of shell plating, forward and aft; gunwale bars; margin plate angles; edges and butts of bulkhead plating..... 4 1-2 "
7. Keelson angles; water-tight bulkhead frames; butts and edges of mast plates; deck plating to beams where single flange beams are fitted to alternate frames; rudder plates; stems; stern posts; keels..... 5 "
8. Frame shell rivets in flat of bottom plating before 3-5 length in vessels of full form forward ..... 5 1-2 "
9. Frame shell rivets in deep ballast tanks above inner bottom, and in fore-aft peak's water ballast tanks. Frame shell rivets for frame spacing of 26 inches or over..... 6 "
10. Frames; reverse frames; floor keelsons; beams angles; deck and hold stringer angles; face bars on web frames and side stringers; bulkhead stiffeners; longitudinal angles on continuous girders; vertical angles connecting floors and girders; deck plating to beams, except where single flange beams are fitted to alternate frames..... 7 "
11. Edge of deck plating..... 4 to 4 1-2 "
12. Rivets are not to be placed nearer edge of plates or angles than their own diameter.

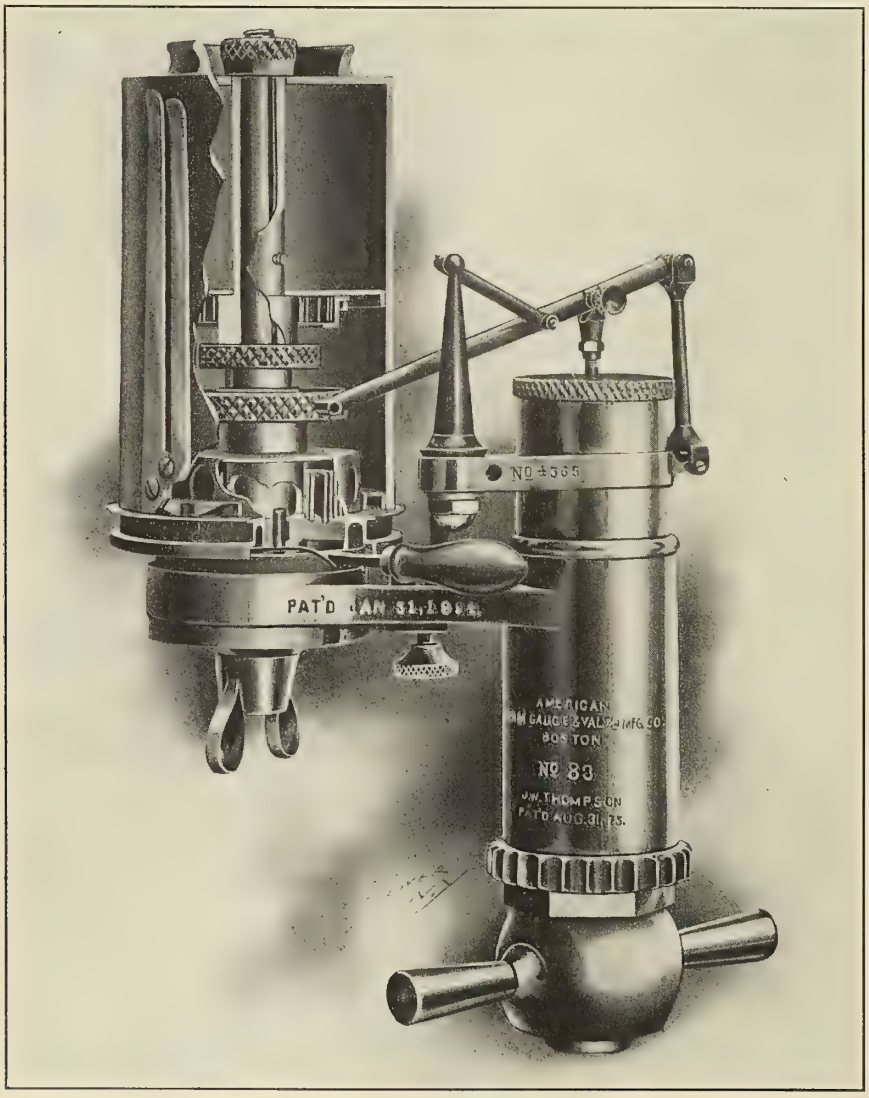
TABLE 19.—LLOYD'S REGISTER. WIDTH OF BUTT LAPS AND BUTT STRAPS. SPACING OF RIVETS.

	Weight of Plate. Lbs.	Diameter.	10–12	12–20	20–26	26–34	34–40
	Diameter of Rivet. Inches.	Pitch.	5/8	3/4	7/8	1	1 1/8
Widths of:	Single riveted butt lap	...	...	...	...	...	...
	Double " " "	...	4 1/4	5	6	...	...
	Triple " " "	...	...	7 1/2	9	10 1/2	12
	Quadruple " " "	...	...	...	12	14	16
	Double " " strap	...	8	9 3/4	11 3/4	...	...
	Treble " " "	...	...	14 3/4	16 3/4	19	21 1/2
	Single " seam lap	...	2 1/4	2 1/2	3	...	...
	Double " " "	6	3 3/4	4 1/2	5 1/4	6	6 3/4
	Triple " " "	...	...	...	7 1/2	8 1/2	9 1/4
	Single " edge strip	...	...	...	...	...	...
	Double " " "	...	...	...	...	...	...
Spacing of rivets.	1.—ϕ of rivet from edge of plate. Edge riveting: ϕ to ϕ of rows....	1 1/2	1 5/16	1 1/8	1 5/16	1 1/2	1 11/16
	2.—Butt straps. ϕ to ϕ of rows.....	2	1 1/4	1 1/4	1 1/4	2	2 1/4
	3.— " laps. ϕ to ϕ of rows.....	2 1/2	1 9/16	1 7/8	2 3/16	2 1/2	2 13/16
	4.—	3	2	2 1/4	2 5/8	3	3 3/8
	5.—Shell butts; upper, middle, spar-deck stringers, except quadruple riveted butt laps.....	3 1/2	2 1/4	2 5/8	3 1/8	3 1/2	4
	6.—Quadruple riveted butt laps; deck and inner bottom plating; hold stringers; floor, tie, and margin plates.....	4	2 1/2	3	3 1/2	4	4 1/2
	7.—Edge af deck plating.....	4 to 4 1/2	...	...	...	...	...
	8.—Long. seams of shell plate forward and aft; gunwale bars; margin plate angles; bulkhead butts and seams.....	4 1/2	2 3/4	3 3/8	4	4 1/2	5 1/16
	9.—Keelson bars; W. T. frames; mast butts and edges; deck plating to beams with single flange; rudder plates; stem, stern post, bar keel.....	5	3 1/8	3 3/4	4 3/8	5	5 5/8
	10.—Frame shell rivets in flat of bottom before 3/8 L, in vessels of full force forward.....	5 1/2	3 7/16	4 1/8	4 13/16	5 1/2	6 3/16
	11.—Shell-frame rivets for frame spacing of 26" or over.....	6	3 3/4	4 1/2	5 1/4	6	6 3/4
	12.—Frames; reverse bars; floors; keelsons; stringer angles; bulkhead stiffeners; floor and keelson clips; continuous girder angles; deck plate to beams with double flange.....	7	4 1/2	5 1/4	6 1/4	7	7 7/8

ENGINEERING SPECIALTIES.

Improved Detent Motion for Indicators.

Ever since the introduction of the steam engine indicator, for the purpose of measuring the power of engines, it has been recognized that a very serious limitation consisted in the fact that the indicator was hardly applicable to fast running engines. This was due mainly to the great difficulty in getting the paper drum connected with the speed-reducing device, and again disconnected for the purpose of removing the cards. This defect has been remedied by what is known as a *detent* motion, applied to the new American Thompson indicator, manufactured by the American Steam Gauge and Valve Manufacturing Company, of Boston, Mass. In the new form the drum carriage, after it is once connected with the reducing motion, need not be disconnected until desired, as means is provided for stopping the motion of the drum without making any change in the connection with the reducing device.



INDICATOR WITH DETENT MOTION DEVICE.

We show herewith an illustration of one of these indicators fitted with the detent motion, and would call attention to the handle shown in the middle of the engraving. By moving this handle, or lever, in the direction traveled by the paper drum, until the drum releases itself, the connection between the drum and the motion of the reducing gear is broken, and a new card may be placed on the drum, or the old one taken off, with great ease. To reconnect the drum with the reducing motion, all that is necessary is to turn the drum by means of the milled rim on the top until the catch draws it into play again. That this form of indicator will be a great boon to those who are indicating engines of either the marine or the locomotive type, and, indeed, of high-speed engines generally, needs no emphasis.

Roller Bushings.—One notable departure in the outfit of the five lightships building at the yards of the New York Shipbuilding Company, Camden, N. J., has been in the roller bushing supplied in their outfit of tackles. It has been the custom of the United States government for some 12 or 14 years past to insist upon the use of a bushing manufactured expressly for its own use at a considerable expense. In equipping these ships, however, a bushing such as is used in the merchant marine has been substituted, and it is believed by many that the innovation is not only in the interests of economy, but efficiency as well.

New Boiler-Scale Destroyer.

The attention of interested persons is being directed to a new patent water softener, exploited by the Hull Boiler-Fluid Company, of Cumberland street, Hull. It is described as a fluid boiler composition which will "soften" any kind of hard water and greatly prolong the life of steam boilers. It prevents scaling, and in the case of boilers that have already become foul it will gradually disintegrate and remove old scale and prevent priming. Samples of old scale are exhibited which the fluid is credited with having broken up. In some cases, it is said, the scale was a hard, cement-like formation nearly an inch thick. After a use of the

scale destroyer for about three months it is claimed the deposit in the boilers was in the form of a fine, smooth powder or dust, which could be washed away by a hose pipe and water, instead of an incrustation that had to be laboriously chipped off with hammer and chisel.

The composition is described as of a purely vegetable nature, no chemicals being used in any way in its manufacture, and there is said to be nothing in it to injure the plates or fittings of the boilers, whether of iron, copper, brass, or other metal. In order to thoroughly test its behavior in this respect, a large steam boiler was filled with the crude undiluted fluid and run at 60 pounds pressure for three weeks. It was then examined by a boiler inspector and an insurance company's man and reported to be in perfect condition. It is also claimed that where this softener is employed the steam may be used for the most delicate operations of any manufactory without the slightest ill effect, and that a large number of representative firms have tried it under actual working conditions, and have written, speaking very highly of its effect. It is now under trial at sea for use in marine boilers with salt water.—*Daily Consular Report.*

TECHNICAL PUBLICATIONS.

Naval Fleets in 1904. By Commandant de Balincourt (French Navy). Size 6 by 4 1-2 inches. Pp. 845 + vi. + 34, with about 400 outline figures of ships. Paris: Berger, Levrault & Company. Price presumably 6 francs, as last year.

The absorbing struggle between Japan and Russia has given rise to the manifestation on the part of the public of a great amount of interest in naval matters. So clearly has it been shown that without the command of the sea, assured to Japan by the first two blows of the war, she could have accomplished nothing in Manchuria, by reason of her consequent inability to land troops for the operations there, that information on the subject in question is eagerly sought on all sides. The present work is a continuation of similar volumes of previous years, this being the fourth in the series. It presents, in very condensed form, a vast amount of information about the ships which form the *materiél* of the various navies of other maritime powers. The arrangement of this subject matter is excellent; on each left-hand page is given a cut showing the outline or outboard profile of the ship to be described, as well as one or more sketch deck plans. These show clearly the general arrangement of the armor protection, the positions of the guns, and the general appearance of the vessel. On the opposite, or right-hand, page are given the data relating to dimensions, displacement, power, speed, battery, armor, fuel supply, and various other items making up the characteristics of the ship, or affecting her fighting value. To this are added, in many cases, notes and comments on the vessel's condition, and such brief criticism as the case seems to require, with occasionally a casual comparison with some other similar ship.

Under the several navies the ships are arranged in approximate order of importance and date, while the various countries are arranged in alphabetical order, each navy being accorded a short preface, outlining the present general tendency of design and construction. As a reference book, which is the primary intention, this little work seems admirably adapted for its purpose; and for those who are conversant with the French language it is full of valuable information, which appears to be correct, and to have been collaborated with great industry and care. Even without a reading knowledge of French, a reader might readily obtain such information as he would require, from the clear and lucid diagrams given. It is needless to state, that as ships are now-a-days built largely in "classes," each class comprising from two or three to as many as a dozen ships, all from the same designs, a single cut and description will readily cover an entire class, and hence the 400 cuts represent in reality something like 1,500 ships.

Self-Propelled Vehicles. A practical treatise with illustrations. By J. E. Homans, A.M. 8vo. Pp. 652 + vii. Bound in black vellum, gilt top, gold titles. 1904. New York: Theodore Audel & Company. Price $2.00.

In presenting the new edition of this work, the publishers announce that the book has been thoroughly revised, and in large part rewritten. There is a vast amount of useful information packed into its 652 pages, and it is so well arranged and so clearly stated that the reader cannot fail to find and comprehend the information given.

The general principles of automobile construction and operation, including steering devices, underframes, wheels, tires, bearings, lubricators, are included in the opening chapters. Then follows an exhaustive account of the theory, construction and operation of gas engines, occupying over 100 pages. Several typical engines are taken up and discussed separately, and their properties, as regards balance, speed and power, are discussed in the light of fundamental principles. The explanations of the governing devices are clear and valuable, while the discussion of ignition, including the hot tube, and the primary and secondary sparks, cannot fail to prove of the utmost value.

Probably the most interesting feature of the entire work is the extensive chapter devoted to the description of leading types of gasoline vehicles, including the most important of American build. In this chapter the reader is informed as to the details of the transmission and control apparatus in each case. The chapters on electric vehicles are also full, and certain to prove of practical use to the owner and chauffeur. Taking the subject of electrical apparatus from the fundamental principles of circuits and batteries, the discussion passes to the theory and operation of generators and motors; the laws of motor operation; the laws involved in computations of speed and power, and the varieties of motor suited to road carriages. Electricity meters are described and illustrated in a brief chapter, and the principles underlying storage batteries, their construction and care, are outlined. All necessary information is given, and the merits of several types of steam carriage are fully set forth.

The book closes with a chapter on "Gasoline Vehicle Management," excellent for its completeness and "useful hints"; another on gasoline cycles that covers the general principles involved in this type of motor.

As to the method of presentation, one remark is in place: since the advent of the motor carriage has created a widespread interest in matters mechanical, bringing many persons who lack previous acquaintance with the mysteries of engine construction and operation into intimate daily contact with practical problems and situations, it is essential that such a treatise as the present one should give the facts with as few technical terms as possible. In this respect the best book on the subject is somewhat like the best automobile carriage—the simplest.

An exceedingly full index at the close of the book puts its contents into "ready reference" shape, an advantage of no small importance in view of contingencies sometimes happening in the use of the 'mobile.

Record of American and Foreign Shipping. Pp. 1,294. Size 8 by 9 1-2 inches. Price $5.00. American Bureau of Shipping, 66 Beaver street, New York.

The volume for 1905 of the "Record of American and Foreign Shipping" (American Lloyd's) which is the thirty-seventh annual issue of this valuable register is now being delivered to subscribers.

The record contains full reports and particulars of about 17,000 vessels ranging from the infrequent ketch to the full-powered transatlantic liner and flying the flag of every nation, alphabetically arranged with much detail as to build, ownership, and condition. This information forms the bulk of the annual volume but it also contains rules for the construction and classification of all classes of vessels with illustrations and tables all of utmost technical and practical value; revised rules for the

construction of machinery and boilers, electric installation and refrigerating apparatus on shipboard.

The volume contains names of vessels which have been changed; list of compound names arranged alphabetically by last word of name for ready reference; list of addresses of prominent shipbuilders, dry-docks and marine railways of the United States; list of owners of vessels, all of much value to the shipping interests. The work is approved and endorsed by the important boards of underwriters of the United States, and is accepted by the merchants and underwriters throughout the world as a standard classification of shipping. Supplements to the volume issued semi-monthly keep subscribers informed of new vessels constructed during the year, with reports of repairs to old vessels, and other useful information.

Mechanical Appliances, Mechanical Movements, and Novelties of Construction. By Gardner D. Hiscox, M.E. Pages 396. Illustrations 970. Price $3.00, bound in cloth. New York: The Norman W. Henley Publishing Company. 1904.

This book, while complete in itself, is in fact a continuation of the author's "Mechanical Movements, Powers and Devices," which has now run into its tenth edition. The aim has been to present to readers information regarding nearly all conceivable devices for producing motion or accomplishing mechanical results. The machines incorporated cover an immense field, and have been carefully selected to supply the needs of the student seeking either general or special information. They will be found fairly representative of the power devices of the modern industrial world, as well as many of those now generally superseded by later and more efficient substitutes. From the multitude of devices described (and in every case illustrated), might be mentioned, in passing, such items as conveyors and elevators, prony brakes, thermometers, various types of boilers, solar engines, oil-fuel burners, condensers, evaporators, Corliss and other valve gears, governors, gas engines, water motors of various descriptions, air ships, motors and dynamos, automobiles and motor bicycles, railway block signals, car couplers, link and gear motions, ball bearings, breech block mechanism for heavy guns, and a large accumulation of others of equal importance. The descriptions are terse and brief to a degree, but are clear and well cover the points to be made. Taken in conjunction with the accompanying illustrations, they will readily afford a splendid idea of what the author intended to convey.

One chapter or "section" is devoted to "Navigation, Vessels, Marine Appliances, etc." Beginning with curious forms of boats used by the North American Indians and the South Sea islanders, the reader is carried through descriptions of the forms of racing yachts, various types of screw propellers, catamarans, steering gears, automobile torpedoes, submarine boats, knots, splices and hitches, buoys, and many other things nautical.

The last section is devoted to the inventions of the multitudinous followers of the Keely motor school:

"Although the author has not the slightest desire to encourage the hopeless pursuit of perpetual motion, he has, nevertheless, thought it advisable to dwell at some length on the exceedingly ingenious means devised by misguided inventors in their endeavors to solve an unsolvable problem. The pages in which perpetual motion machines are described may induce those who still believe in this *ignis fatuus* to bend their energies in causes more worthy of their zeal. Moreover, it may be that some of the mechanical movements which have been evolved by the perpetual motion inventor, although they may not attain the end sought by him, may still be applied with profit to his instruction in true mechanical principles and to avoid the errors committed in the search on the lines of this folly of past centuries. This in itself is a sufficient justification for the insertion in this volume of the section on perpetual motion."

Typograpy is excellent; cuts clear and well drawn. The work may be heartily recommended to inventors, patent attorneys, machinists, engineers and all others engaged in the mechanic arts, or interested in the design of any form of machinery.

Gruppeneinteilung fuer die Gewicht und Kostenberechnung von Schiffen. By F. Meyer. Size 4 1-2 by 6 1-2 inches. Pp. 52. M. Driesner, Klosterstr. 45, Berlin, Germany. Price $1.25.

This pamphlet contains a list of names of all parts entering into the construction of a merchant ship. The book is supposed to be a help in calculating the weights, as the author states that frequently a number of small parts are omitted in weight calculations, causing serious errors in displacement and estimated price. The author has endeavored to classify all the material, although he admits incompleteness, as each shipyard has a different method of calculation. A table is also included which shows a system of tabulating the cost of the different parts.

A Modern Battleship. New York: The Derry-Collard Company; and London: The Locomotive Publishing Company. Price 50 cents; edition-de-luxe, framed, $5.00.

This is an engraving 36 inches long, on plate paper 44 by 28 inches, of the inboard profile of a battleship of the *Connecticut* class, showing in great detail the contents of the various compartments and the multitudinous appliances obtaining on a ship of this character. Each item is numbered, and the index to the nearly 500 parts is printed below the engraving. The value of such a pictorial description for conveying accurate information to those who may require it can hardly be overestimated.

The "Mechanical World" Pocket Diary and Year Book for 1905. Size 4 by 6 1-4 inches. Pages 391. Figures 66. Manchester: Emmott & Co., Limited. Price six-pence, net.

Within a small compass, in a book conveniently carried in the pocket, have been collected a large number of tables of engineering data of all descriptions, from the usual mathematical and trigonometrical tables to horsepower, steam and vacuum tables, tables of the properties of I- and Z-bars, shafting and the strength of materials, electrical constants and wiring tables, hydraulic data, tables of bolts, nuts and threads, conversion tables between metric measures and the corresponding British units—in short, all of the usual data and tables to be found in the general engineering reference books. In addition to this are to be found many "chapters" or notes on various subjects of an engineering character, such as engines, boilers, valve setting, pumps, oil and gas engines, belt and rope driving, electric machinery, power transmission and devices, and a multitude of other items of interest to the engineer. One of the notes which will appeal with particular emphasis to our readers, is on the speed and power of small launches, giving, among other things, three tables: values of $D^{\frac{2}{3}}$ for displacements from 500 to 24,000 pounds, where D is expressed in gross tons; values of S^3 for speeds from 5 to 19.5 miles per hour, where S is expressed in miles per hour; and values of C, the Admiralty coefficient, for a special case, at speeds from 7 to 15 miles per hour.

In the rear of the book are a diary and blank pages for memoranda. A splendid and very complete index renders the book exceedingly easy of access, and adds enormously to its value as a work of ready reference.

QUERIES AND ANSWERS.

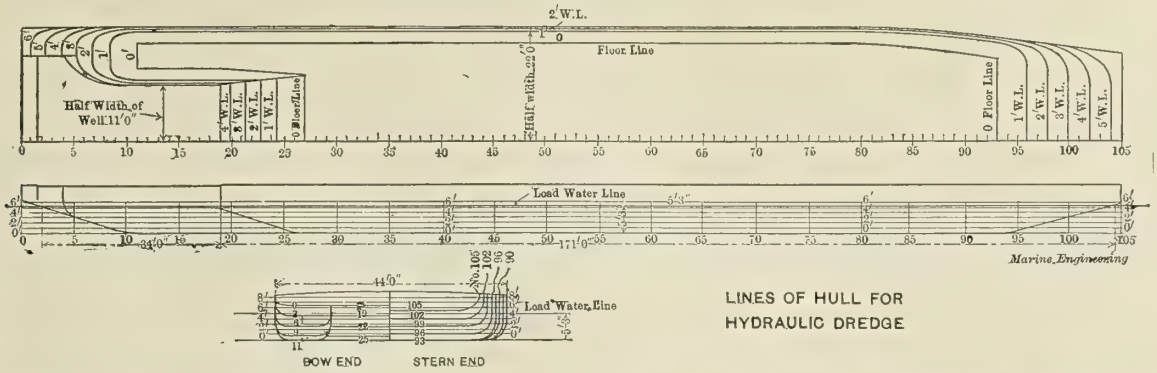
All reasonable questions concerning marine engineering received from and signed by subscribers will be answered by the editor in this column. All communications must bear the name and address of the writer.

Q. 265.—Will you kindly answer the following queries in the columns of your valued paper?

1. What I. H. P. would you advise putting into a vessel of the dimensions and forms indicated on the inclosed blue print, in order to obtain a speed of 10 knots per hour in still water? The area of midship section is 227 square feet; the wetted surface is approximately 9,680 square feet; and the displacement at load water line is 1,050 gross tons.

2. What rules are applicable to such cases as this, by the use of which the horse power required can be determined with some hope of obtaining results on trial, that will correspond with the calculations? X. M. T.

A.—The indicated horsepower required for propelling a ship



LINES OF HULL FOR HYDRAULIC DREDGE

at any stated speed is given by the expression:

$$I.H.P. = \frac{D^{\frac{2}{3}}S^3}{C},$$

where D is the displacement in tons, S the speed in knots, and C a coefficient known as the Admiralty coefficient, and varying with the form of the hull, the size of the ship, and the speed desired. In the present case it is not at all certain what value of coefficient will be applicable, because the form of the hull is very unusual for anything but very low speed. Assuming, however, that the value may be placed at 149, we find that the horsepower will be represented by the expression:

$$\frac{103.3 \times 1{,}000}{140} = 738.$$

It thus follows that if an engine of 750 horsepower be fitted to the hull, it is quite reasonable to expect that the speed of 10 knots might be realized.

Q. 270.—Kindly inform me in your next issue of MARINE ENGINEERING. (1) Which would pull the most at the greatest speed, and why, the *Kaiser Wilhelm der Grosse* of 28,000 horsepower, or 28 tugs of 1,000 horsepower each?

(2) What process can I use to figure out the area of a funnel?

J. L. D.

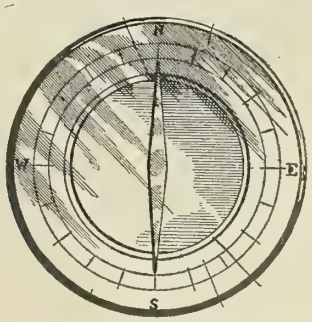
A.—Twenty-eight tugs of 1,000 horsepower each could tow a greater weight than could the *Kaiser Wilhelm der Grosse* of 28,000 horsepower, for the simple reason that the tugs were designed to develop 1,000 horsepower each at the speed at which they would tow, whereas the *Kaiser Wilhelm der Grosse,* when the engines are developing 28,000 horsepower, is running at full speed, and all of the power is required in overcoming friction and the ship's own resistance, there being nothing left for towing.

(2) The cross sectional area of a funnel is easily figured out from the grate surface, the proportion being 1 square foot in the funnel to every 6 or 8 square feet of grate area which the funnel serves.

SELECTED MARINE PATENTS.

774,693. STEERING APPARATUS FOR SHIPS. JOHN PETERSON, ABENIA, N. Y.

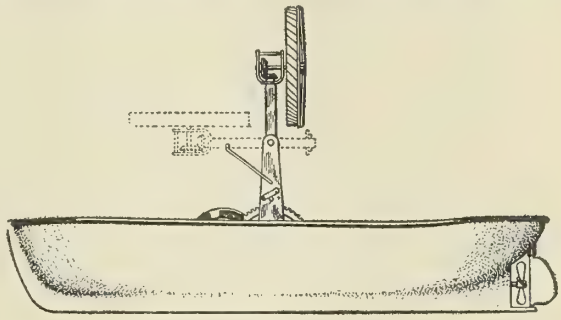
Claim.—5. The combination of a compass having an oscillatory element disposed concentrically beneath its magnetic needle, electrical conductor



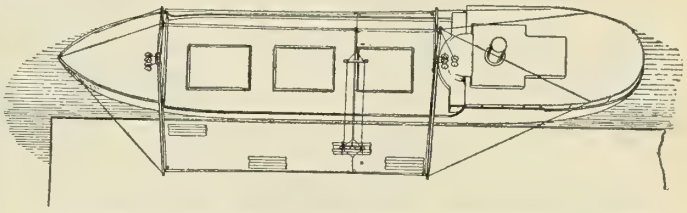
rails upon said movable element, an electrical generator, electrical conductors connecting with electrical generator, and conducting rails in open circuit, a circuit-closing device carried by the magnetic needle of said compass and coacting with said conductor rails, means for oscillating said oscillatory element, and a set-screw for securing said oscillatory element in an adjusted position, substantially as described. Nine claims.

775,971. BOAT PROPELLER. HERMAN HELMEKE, ST. LOUIS, MO.

Claim.—3. In a boat, suitable standards, a drive shaft mounted on the same, a propeller shaft, intermediate gearing between the propeller shaft and drive shaft, a gear wheel on the drive shaft, a tilting frame mounted to swing vertically on the standards, a shaft disposed longitudinally in said frame, and adapted to have one end co-operatively connected to the afore-



said gear wheel, a rotatable sleeve enveloping the shaft on the frame, an upper terminal fork carried by said sleeve, a wind wheel shaft on the fork, a bevel pinion on the shaft, a corresponding pinion on the upper end of the longitudinal shaft meshing with the aforesaid pinion, means for rotating the sleeve to effect the proper angular adjustment for the wind wheel, and means for locking the sleeve when once adjusted, substantially as set forth. Seven claims.



777,122. HOISTING AND CONVEYING APPARATUS. THOMAS S. MILLER, SOUTH ORANGE, N. J.

Claim.—15. In a boat, in combination, two masts, booms extending outboard therefrom, a hoisting rope, a swinger rope engaging therewith, a hoisting rope sheave traveling between said masts, a swinger rope sheave traveling between said booms, a traction actuator connected with both of said sheaves, a swinger actuator and a hoist actuator; each of said actuators being located adjacent to one of said masts. Twenty-one claims.

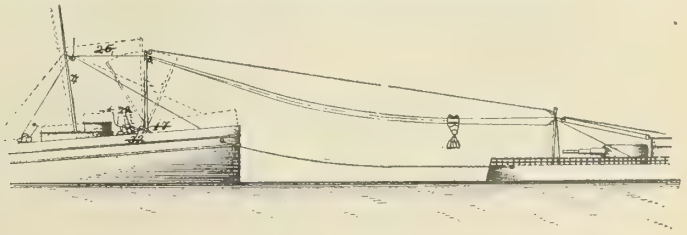
777,619.—DEVICE FOR USE IN HYDRAULIC ENGINEERING. DANIEL J. FEUERLOSCHER, GRATZ, AUSTRIA-HUNGARY.

Claim.—1. The device consists of a body formed by inclosing concrete in an outer casing or envelop formed of boards, battens, or laths lined with sack-cloth or the like and tied at suitable intervals throughout its length, similarly to a fascine, and which being sunk in the water before the concrete has set will adapt itself to the configuration of the bottom, thus preventing underwashing or undermining of the base of foundation. Four claims.

777,685. CONVEYING APPARATUS. THOMAS S. MILLER, SOUTH ORANGE, N. J.

Claim.—1. In combination, a boat, an elevated support thereon, a rope extending from said support to another support located off of said boat, a load carriage traveling to and from said boat on said rope and a means for maintaining a normal elevation of said load carriage during its excursions, which means consists of mechanism whereby the yielding power of a fluid is applied in opposition to the strain on said rope, substantially as described.

2. In a conveying device the combination with two relatively movable bodies each carrying a cableway support, one of said supports containing a hinged member, a cable extending from said hinged member to the other support, means for producing tension on said cable acting through the



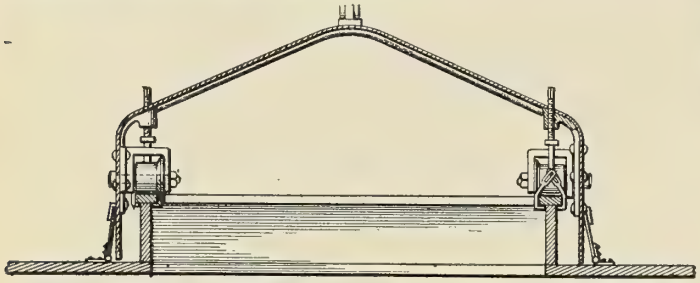
hinged member, and means for transferring loads over said cable comprising ropes leading substantiallly through the pivot-axes of the said hinged member.

5. In combination two boats, a cable extending between said boats, a hinged support for said cable on one boat, an engine connected with said hinged support to produce tension on the said cable, a carriage traveling on said cable, a traversing rope for said carriage, and guides for said traversing rope located to substantially coincide with the pivot-axes of the hinged support.

16. Apparatus for transferring cargoes from one vessel to another comprising an anchorage to a fixed support on one vessel and a tower oscillating from its point of support on the other vessel, a traveling cable passing from one anchorage to the other and return, a pulley on one of said anchorages around which said cable passes and means to maintain a constant tension on each portion of said cable. Sixteen claims.

777,878. HATCH FOR SHIPS. EARL C. AKERS, PORT HURON, MICHIGAN.

Claim.—1. The combination with a ship's deck and a hatchway therein, of coamings rising from the deck, a hatch cover comprising a plurality of cover sections which are joined end to end by water-excluding joints and provided with rollers which rest on said coamings and are adapted to roll thereon from one end of the hatch to the other, and means for separately fastening the cover sections to the coamings.

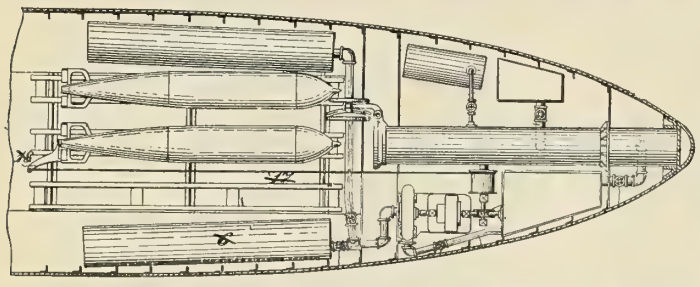


4. The combination with a ship's deck and a hatchway therein, of coamings rising from the deck, a hatch cover comprising a plurality of cover sections having at their sides and ends depending portions located laterally outside of the coamings, means for supporting said cover sections on the coamings, means for providing substantially water-tight joints between the meeting margins of the sections, plates hinged to the deck and adapted to swing upwardly against the said depending parts of the cover sections and latches for fastening the hinged plates in their upper position.

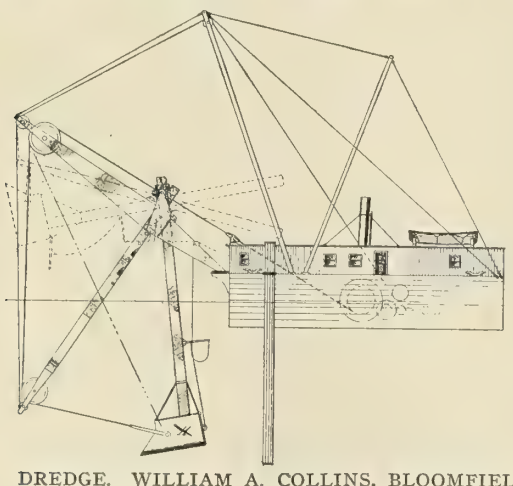
15. The combination with a ship's deck and a hatchway therein, of coamings rising from the deck, a hatch cover extending across and supported on said coamings comprising a plurality of cover sections which are transversely corrugated, the meeting margins of adjoining sections lapping one over the other with the corrugations meeting in interfitting relation, means on the adjacent margins of the cover sections for positively locking the same together and means for exerting tension on the interlocked cover sections longitudinally of the hatchway. Seventeen claims.

778,339. COMPENSATING DEVICE FOR SUBMARINE BOATS. LAWRENCE Y. SPEAR, GREENPORT, N. Y., ASSIGNOR TO ELECTRIC BOAT CO., N. Y., A CORPORATION OF NEW JERSEY.

Claim.—1. This invention relates to the class of submergible and submarine boats which carry torpedoes and have means for expelling them; and it has for its object improved means for compensating the weight of



torpedoes moved in a fore and aft direction within the boat with the purpose in view of maintaining constant or very nearly constant the fore and aft center of gravity of the system at all times during the shifting and after the shifting of the torpedo. Two claims.



778,634. DREDGE. WILLIAM A. COLLINS, BLOOMFIELD, N. J.

Claim.—4. In a dredge or the like, the combination with a boom, a dipper and a dipper handle, of an operating chain secured to the dipper and passing over the outboard end of the boom, means for operating said chain, and a radial arm mounted on the boom and interposed on the operating chain between the dipper and the outboard end of the boom, said arm being adapted to regulate the working angle of the operating chain.

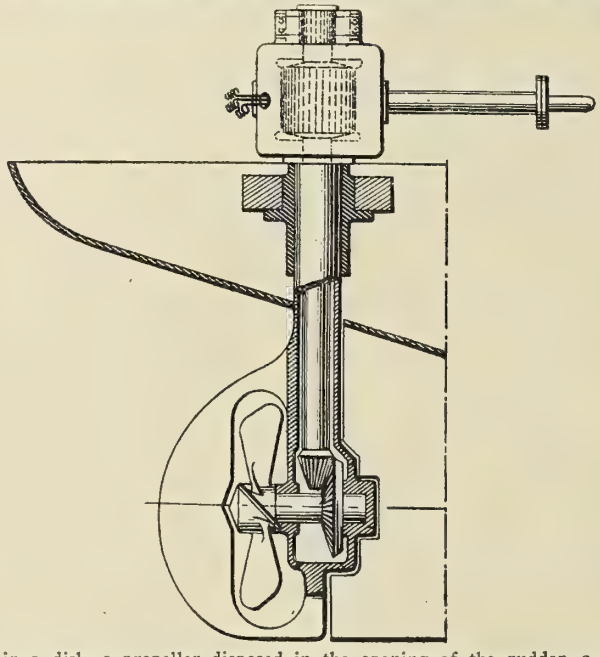
39. In a dredge or the like, a boom, a support therefor, means for swinging said boom in a horizontal plane, a dipper, a dipper handle, carried by the boom and swingable in a vertical plane, an operating chain secured to the dipper and rove through a sheave at the outboard end of the boom, means for operating said chain, a radial arm swingable in a vertical plane and interposed on the operating chain between the dipper and the outboard end of the boom adapted to regulate the working angle of the operating chain, a rope secured to the outer end of the arm and adapted to limit the descent of said arm, a slack-take-up for said rope, and a stop adapted to limit the inboard movement of the radial arm.

50. In a dredge of the character set forth, a radial arm substantially oblong in cross section and having its broader sides facing inboard and outboard.

51. In a dredge, a boom, a dipper handle, and means for operating the same and an adjustable stop on the upper end of the dipper handle. Fifty-five claims.

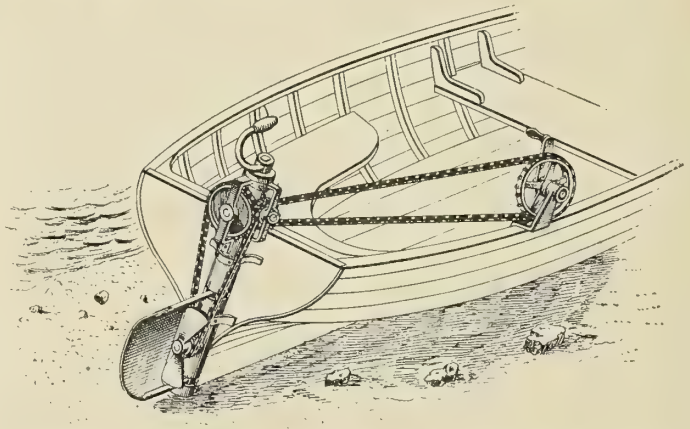
778,435. BOAT PROPELLER. GEORGE F. ATWOOD, WAKEFIELD, MASS., ASSIGNOR OF ONE-HALF TO HERBERT C. HALL, TRUSTEE.

Claim.—1. The combination of a boat, a vertically disposed sleeve or hollow post journaled in a bearing on the boat, a rudder carried by the sleeve or post and having an opening, a casing arranged on the upper end of the post, and having a hollow handle terminating at its outer



end in a disk, a propeller disposed in the opening of the rudder, a shaft carrying the propeller and journaled in the lower portion of the sleeve or post, and provided within the latter with a beveled gear, an upright shaft journaled in the sleeve or hollow post, and provided with a beveled gear intermeshed with that on the propeller shaft, an electric motor comprising a rotary armature fixed directly on the upright shaft, a source of electric energy, a hand grasp pivotally connected to the outer end of the hollow handle whereby it is adapted to turn on its axis, and having a disk opposed to that of the handle, and coacting means on the disks for controlling the supply of electric energy to the motor. Four claims.

778,763. BOAT PROPELLER. JOHN SALOM, NEW YORK, N. Y., ASSIGNOR OF ONE-THIRD TO JOHN F. CUNNINGHAM, NEW YORK, N. Y.



Claim.—1. The principal feature of the invention lies in a certain novel arrangement which enables the device to be easily and quickly applied to or removed from the transom of a boat. This adapts the mechanism particularly to boats which are to be launched frequently, and the principal advantage lies in the fact that when the boat is hauled up on the shore or in the davits of a vessel the propelling device may be removed from the transom and stowed safely away. Seven claims.